Lloyd's Register Technical Association

INERT GAS AND VENTING SYSTEMS

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Written contributions to the discussion of this paper are invited from members of the Lloyd's Register Technical Association. To ensure inclusion in the discussion paper, the contributions should be received by the Hon. Secretary in London not later than the 31st December, 1986.

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by

B. W. Oxford

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B. W. Oxford served his apprenticeship with Richardsons and Westgarth, Hartlepool, and the steering gear manufacturers Brown Brothers in Leith. He obtained his seagoing experience with Shaw Savill Line before taking up duties with the Society at Hull in 1956 and later serving at various U.K. locations and overseas in Aden and Bergen. Since 1970 he has been in HQ (MDAPAD) dealing with plan appraisal of pumping and piping aspects for ships and was appointed Principal Surveyor in January 1986. He attends the IMO Sub-Committee on Fire Protection on behalf of a national delegation and also advises IACS delegates on pumping and piping aspects.

INTRODUCTION

Since inert gas and venting systems have been passing through an evolutionary period over the last few years, with new or updated Rules and Regulations resulting in modified systems and changes of application, it is the Author's intention to keep this paper as simple as possible and to concentrate on those aspects of the Rules and Regulations of the Society and other regulatory bodies which, it is hoped, will be of most benefit to the Society's surveyors. In particular, many references will be made to the Regulations of IMO (International Maritime Organisation) known before 22nd May, 1982, as IMCO (International Governmental Maritime Consultative Organisation).

By making such frequent references to IMO it may be felt that the Society is losing part of its identity. However, since members of the Society attend IMO on a regular basis and take part in the working parties and discussion groups on many subjects, the Society's influence is still felt throughout the world in the written word that eventually appears in the form of the famous IMO blue books. Certainly, from plan approval aspects, it makes life easier to know that when IMO Regulations appear, word for word, in the Society's Rules as, for example, in the gas carrier and chemical tanker Rules, the person at the other end of a letter, telex or telephone is looking at and quoting from exactly the same text. There is still plenty of room for interpretation and comment as is in evidence by the additional comments shown, for example, in the Society's Rules for Liquefied Gas Ships. It will also be well known that the Society's Rules are written for *new* ships only and whilst these may be used for guidance in their application to existing ships it has proved useful to the Society's surveyors that, at least in respect of inert gas systems, IMO has, for the first time, detailed the relaxations which may be given for existing ships depending on the date when the inert gas system was first fitted.

As suggested by the title it is proposed to deal with this paper in two separate parts.

PART I

INERT GAS SYSTEMS

HISTORY

Inert gas, developed from boiler flue gases, was first used as a means of preventing explosions within the cargo tanks of crude oil tankers by an American tanker company in 1925. This was later abandoned due to difficulties caused by the short voyages of the ships concerned but re-introduced by another American company around 1932, again as a safety measure.

It was subsequently realised that having an inert gas system in operation reduced corrosion in the tanks and that the pressure exerted by the inert gas on top of the oil helped to reduce the time taken in port to discharge the cargo.

It was not until the early 1960's that other major oil companies began to take an interest in the benefits of fitting inert gas systems especially with regard to protecting cargo tanks against explosions.
However, towards the end of 1969, explosions on board three very large crude oil tankers, the "Marpessa", "Mactra" and "Kong Haakon VII", all of which were cleaning empty cargo tanks but not operating with inert gas at the time, set in motion extensive investigations into the cause or causes of these explosions. It is to the great credit of the tanker companies concerned and to the various bodies and testing facilities throughout the world that such great effort was put into finding the possible causes of these explosions and the findings have greatly enhanced the safety of similar ships.

Without going into too great detail of these investigations, consideration was first given to the following aspects as representing possible sources of ignition within cargo tanks:

(a) Generation of electrostatically charged mist from tank washing machines.

(b) Mechanical impact by falling bodies such as anodes.

(c) Pyrophoric ignition (see paragraph 12.6 of Guidelines for Inert Gas Systems in IMO blue book entitled "Inert Gas Systems 1983 Edition" Sales No. 860.83.15.E).

(d) Auto-ignition from hot steam coils.

(e) Ignition by gas compression due to sloshing.

(f) Sparks induced by radio frequency generating apparatus such as aerials.

Subsequent investigations showed that aspects (b) to (f) were unlikely to be the cause of these explosions and that the generation of an incendive spark caused by the electrical discharge from a slug of water from a high capacity tank washing machine (above 60 m^3/hour) charged up by its passage through an electrically charged mist, was the most likely culprit. The practice of re-circulating the washing water in closed circuit via the slop tanks and the use of chemical additives were also considered to be contributory factors in increasing the electrical charges developed.

Therefore in 1971 IMCO, as it was then called, adopted "Recommendations on Fire and Safety Requirements for the Construction and Equipment of Tankers". These recommendations were later included in SOLAS 1974 and, as far as inert gas systems were concerned, resulted in Regulation 62 of Chapter II-2 which, for the first time, addressed itself to the purpose, design and operational details of an inert gas system and which, above all, stated that "the inert gas system should be capable of providing on demand a gas or mixture of gases to the cargo tanks so deficient in oxygen that the atmosphere in a tank may be rendered inert, i.e. incapable of propagating a flame". It should be noted, however,

that, at that time, this new regulation applied to *new* tankers only and was applicable only to new tankers of 100,000 metric tons deadweight and upwards and combination carriers of 50,000 metric tons deadweight and upwards.

The tonnages were no doubt based on the presumption at the time that tankers and carriers of lesser tonnage would not be utilising high capacity tank washing machines.

Regulation 62 therefore formed the basis for the design of inert gas systems for several years until 1978 when IMCO convened the International Conference on Tanker Safety and Pollution Prevention. This conference was convened as a matter of urgency on account of the increasing number of ship casualties especially around that period due to various causes and was charged with the task of setting up further studies to improve the safety of ships and property at sea having particular regard to tankers and the lives of persons on board.

This resulted in what is described as the 1978 Protocol which concerned itself with amendments and modifications, firstly to SOLAS 1974 and secondly to MARPOL 1973. One of the regulations so affected was Regulation 60 of SOLAS Chapter II-2 which, as amended, required that all *new* tankers above 20,000 metric tons deadweight would now require to be fitted with inert gas systems for the internal protection of cargo tanks and in addition that such inert gas systems should be in operation during crude oil washing of the cargo tanks. From the foregoing it would appear that no distinction was now being made between tankers and combination carriers nor between tankers carrying oil, chemicals or liquefied gases. However, since the application of inert gas systems on chemical tankers is a story in itself, this aspect will be dealt with separately later on.

Returning to the 20,000 deadweight figure this meant that a far greater number of tankers than previously were going to require to be fitted with inert gas systems and IMCO therefore decided that the time was opportune to revise the inert gas requirements with the result that Regulation 62 was revised, three times to be exact, and now appears, still as Regulation 62 in Chapter II-2, in the 1981 Amendments to SOLAS 1974 (Sales No. 09282.01.E) which came into effect for both *new* and *existing* tankers on 1st September, 1984. One might have thought that this would have been the final form of Regulation 62 but this was amended yet again by Volume 1 of the 1983 Amendments to SOLAS 1974 affecting, fortunately and quite rightly, only Regulation 62.20 concerning the venting requirements on *existing* tankers.

At the same time as revising Regulation 62, IMCO also published "Guidelines for Inert Gas Systems" and these now appear in an IMO blue book entitled "Inert Gas Systems 1983 Edition" (Sales No. 860.83.15.E).

The guidelines themselves contain a lot of information and are a very useful guide in dealing with the details of an inert gas system; its purpose; its design in general and for particular items of equipment; interpretation of regulations; operational aspects and the requirements for maintenance and emergency procedures. From time to time reference will be made to these Guidelines as appropriate.

So far only IMCO's , or IMO's, involvement with inert gas systems has been mentioned and one may wonder where Lloyd's Register of Shipping enters the picture.

Well, Lloyd's Register has been involved with inert gas systems since 1953. At that time acceptance of a system was based on tanker company experience, good engineering practice and consideration of the safety aspects involved. Then, as a parallel to SOLAS 1974, the Society's first Rules for inert gas systems were published in 1973. These Rules were slightly amended in 1975 and again in 1978 when the Society's Rules were published in an entirely new format.

However, as previously mentioned, since the Society has been closely involved throughout the years at IMO in the working parties developing both Regulation 62 and the inert gas Guidelines it was perhaps inevitable, but also welcome, that the Society's Rules now incorporate those of IMO and appear in Part 5, Chapter 15, Section 7.

The main difference between the Society's Rules and Regulation 62 is that the Society's Rules apply only to *new* tankers, whereas Regulation 62, paragraph 20, also offers some relaxations for *existing* ships. As many surveyors will know, Regulation 62.20 is of particular importance in an exercise presently being carried out by the pumping and piping section of engine plans through the auspices of, and in conjunction with, the International Conventions Department Safety Equipment Section. Regarding this exercise, inert gas systems on board classed ships for which the Society is authorised to issue the Safety Equipment Certificate on behalf of Administrations, are being checked for compliance with Regulation 62 by filling in a questionnaire concerning the requirements which are detailed in Headquarters letter ICD/ICL/124 dated 8th August 1984.

Before concluding this History it may be of advantage to discuss why an inert gas system is required to be in operation when cargo tanks are being washed with crude oil as referred to in Regulation II-2, 60.6 of the 1981 SOLAS Amendments.

Firstly, it should be noted that any size of crude oil tanker, that is both above and below 20,000 metric tons deadweight, which uses crude oil washing shall be fitted with and operate an inert gas system during that process. The reason for this is that, unlike water washing of cargo tanks which was normally carried out at sea during the ballast voyage, crude oil washing is normally required to be carried out in port. This gave concern to some Administrations and Port Authorities that, since the evolution of gases by crude oil washing were estimated to increase seven fold over those by water washing then, without the presence of inert gas, large amounts of flammable gases could be expelled into the atmosphere of the port.

It might also be supposed that, due to the high evolution of gases within the cargo tanks when crude oil washing, the atmosphere within the cargo tanks would be over-rich and therefore non-flammable. However it was discovered in an investigation carried out by a major oil company that despite this high gas evolution factor it was impossible to guarantee that large flammable areas did not exist within the cargo tanks. Finally it was also verified by the same company that tank washing by either seawater by itself, or crude oil by itself, was likely to cause less generation of static electricity than the highest which could be obtained when either 95% crude oil and 5% seawater were mixed to give a positive charge or 25% crude oil and 75% seawater were mixed to give a negative charge as indicated in Fig. 1.

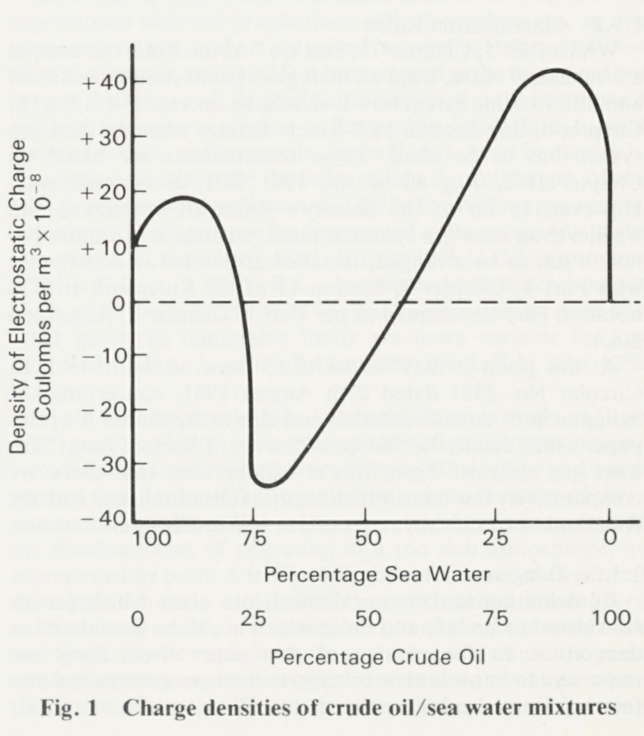


Fig. 1 Charge densities of crude oil/sea water mixtures

It is for the reasons stated above that it is so important for crude oil to be drained from the tank cleaning pipes after crude oil washing and before water rinsing the tanks or to drain water from the lines before crude oil washing.

Finally, as per Regulation 60.7, all tankers fitted with a fixed inert gas system shall be provided with a closed ullage (level gauging) system. This is obviously to prevent the escape of inert gas through manual openings but other operations such as sampling the tank atmosphere for oxygen or hydrocarbon content and dipping after crude oil washing of necessity have to be allowed and inert gas will escape but hopefully for a limited period only.

1.1 General Comments

1.1.1 Plan Approval

So far as Head Office is concerned plans of inert gas systems are approved in the pumping and piping section of the Machinery Design Appraisal and Plan Approval Department, otherwise known as MDAPAD or Engine Plans.

The purpose of this paper is to give some interpretations of the Rules and explain what requires to be looked at when approving plans of inert gas systems. At this stage only inert gas derived from the flue gases of boilers is being considered but reference will be made to other systems later on.

First of all it should be emphasised that only diagrammatic plans of inert gas systems require to be submitted for classification purposes (*See also* 1.1.9). Therefore and inevitably the plans which are appraised bear little relationship to the actual installation as seen by surveyors on board the ship and problems can arise between what the surveyor is faced with and the need, so far as practicable, to comply with the Rules. This is often more evident on existing tankers where an inert gas system may need to be fitted for the first time and there is not always the room to make necessary structural changes or to lead the pipework in the normal manner.

Perhaps the only comfort that can be given in these circumstances is that most plan approval surveyors have also served in the outports and can appreciate such difficulties. Also the Society's Rules are not so rigid that equivalent arrangements cannot be accepted in many instances. It can also be said that feedback from the surveyors on any problem is welcome.

1.1.2 Classification Rules

Whilst Part 5, Chapter 15, Section 7 of the Rules dictates, in a mechanical sense, what an inert gas system should consist of and also to some extent how it should be operated, it is Part 6, Chapter 4, Sub-Section 19.9 which dictates when an inert gas system has to be fitted. These requirements are based on Chapter II–2, Reg 60 of the 1981 SOLAS Amendments. However, so far as the Society's Rules are concerned and whenever an inert gas system is fitted, whether it is required or not, it has to be arranged, installed and tested in accordance with Part 1, Chapter 3, Section 17 of the Rules and an IGS notation may be assigned as per Part 1, Chapter 2, paragraph 2.4.1.

At this point it may be useful to draw attention to L.R. Circular No. 2381 dated 27th August 1981, concerning the assignment of an IGS notation and also to Appendix A of this paper which details the "Scope of Survey of Various Items" for inert gas systems. From this it will be seen that there are comparatively few items which require full compliance with the Rules unless specifically requested by a manufacturer or owner.

1.1.3 Dangerous Areas

By definition tankers are divided into areas which can be considered to be safe and those which are to be considered as dangerous. In the context of this paper these areas are important to know both in relation to inert gas systems and also for cargo tank venting arrangements.

If reference is made to Part 6, Chapter 2–2, Sub-Section 10.10 of the Electrical Rules it can be seen that the dangerous zone is like a blanket over the full breadth of the ship and extending over the length of the cargo oil tanks to a line athwartship 3 metres forward of and 3 metres aft of the foremost and aftermost cargo tank bulkheads and reaches to a height of 2.4 metres above the deck. In theory this blanket also drapes over the deck at each side to the water line at a thickness of 2.4 metres but this has to be disregarded to some extent otherwise tugs or other propelled craft, which are not strictly considered fire safe, could not come alongside.

Since, at a later stage in this paper consideration will also be given to chemical tankers it may be well to mention that for some very dangerous cargoes the 3 metre and 2.4 metre distances mentioned above would both require to be increased to 4.5 metres for compliance with Part 6, Chapter 2–2, Paragraph 12.1.1.

1.1.4 Purpose of Inert Gas

Now to come to the crux of the first part of this paper and pose the question, "Why are inert gas systems fitted?"

The answer is to prevent a flammable mixture from forming above the oil in a cargo tank by reducing the amount of oxygen in the ullage space to less than 11% by volume. Below this percentage a hydrocarbon flame cannot generally be propagated or sustained. In other words it would be snuffed out due to lack of oxygen and an explosion couldn't then occur.

1.1.5 Definition and Cause of Explosion

By definition, an explosion is caused when a source of ignition with sufficient energy to sustain a flame is applied to a flammable atmosphere contained within a tank such that the resulting combustion and expansion of the hot gases take place at such a rate that the gases are unable to escape fast enough such that a pressure is built up within the tank which causes it to fail to the accompaniment of sound and pressure waves which denote an explosion.

From the foregoing it can be deduced that three conditions must be fulfilled in order to give conditions for an explosion and these may be more readily remembered by reference to Fig. 2, where the triangle represents these three conditions.

If any one side of the triangle can be eliminated then the conditions for an explosion no longer exist.

If the hydrocarbon gas is considered first, it is obvious that this cannot be eliminated since it is being generated from the cargo being carried.

Secondly, if the source of ignition is considered, there is no guarantee that this can be eliminated since it can come either from within the tank, caused, say, by static electricity or from outside the tank from another electrical source such as lightning. Other more obvious sources may be incendive sparks caused by fittings dropping within the tanks or smoking on deck.

So this leaves only the third possibility which is to replace the air, which contains 21% oxygen, with a gas containing less than 11% oxygen so that, as previously mentioned, a flame cannot be sustained. In other words, replace the air with an inert gas which will not support combustion.

1.1.6 Flammability

Since air is comprised of 79% nitrogen and 21% oxygen by volume it is not necessary to consider the nitrogen consituent since nitrogen is already an inert gas and is widely used in its vapour form for inerting spaces, for example, on chemical tankers and liquefied gas carriers. However, since it is the oxygen constituent of air which feeds a fire it is now possible to set up a diagram as shown in Fig. 3.

This figure shows how a mixture of hydrocarbon gas and air within a cargo tank and in a flammable state can be rendered safe by replacing the air, which is represented by the oxygen content only, with an inert gas containing say only 5% oxygen.

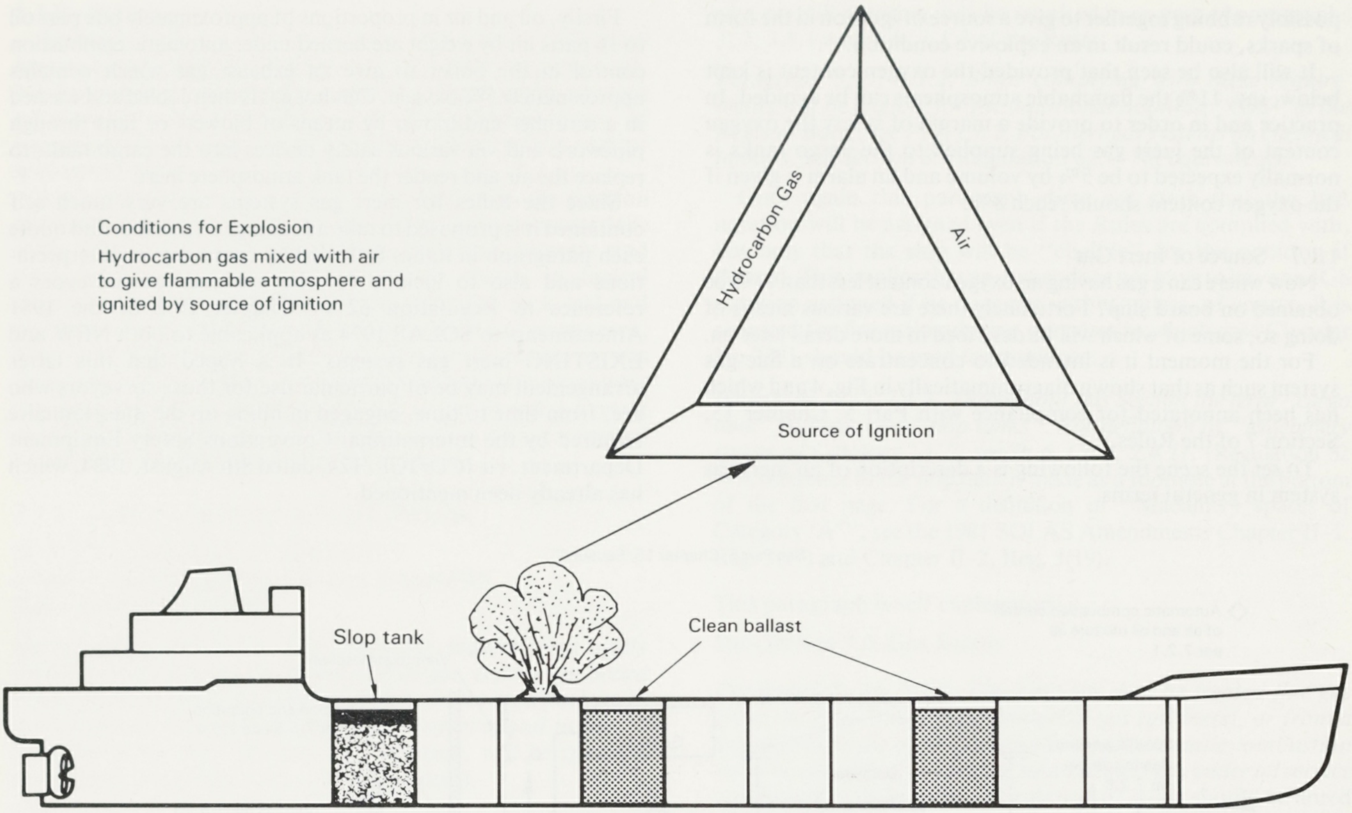


Fig. 2 Conditions for an explosion

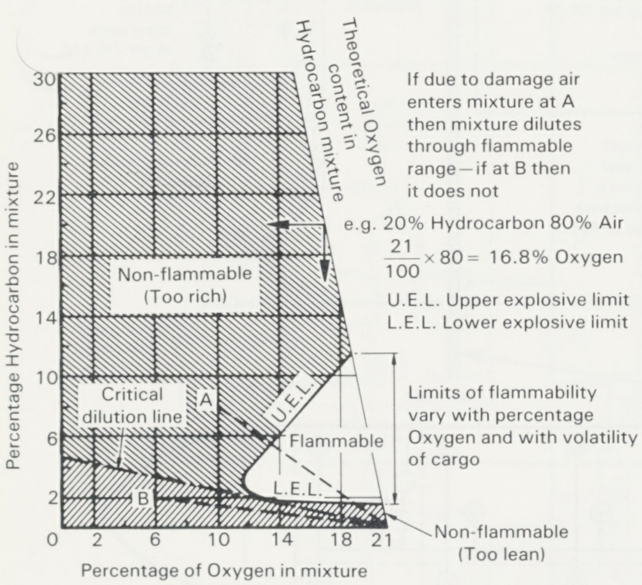


Fig. 3 Limits of flammability of hypothetical crude oil vapour hydrocarbon/air mixture (air 79% nitrogen/21% oxygen)

This is achieved by purging the cargo tank with inert gas and gradually changing the atmosphere by forcing the air out through vent masts, purge pipes and similar openings until the tank atmosphere has been changed several times and then testing the efflux to determine whether or not the tank atmosphere has been inerted.

Referring to Fig. 3 the flammable area is developed from experiments whereby proportions of hydrocarbon gas, in this case crude oil gas, and air are mixed in a vertical tube of not less than 2″ diameter and 5 to 6 feet long, closed at the upper end but open at the lower end. A small flame is applied at the bottom end and by trial upper explosive, or flammable, limits (U.E.L.) and lower explosive, or flammable, limits (L.E.L.) can be established as determined by the flame travelling the full length of the tube. Upper and lower flammable limits for various pure gases, when mixed with air, can be determined quite accurately in the same way e.g., methane 5.3% L.E.L. to 14% U.E.L or propane 2.2% L.E.L. to 9.5% U.E.L., but since the hydrocarbon gas given off by crude oil contains various constituents of these and other gases the flammable limits are more variable but, as indicated in Fig. 3, generally between 2% L.E.L. and 12% U.E.L.

It will also be seen in Fig. 3 that reference is made to "too rich" and "too lean" atmospheres and before inert gas systems were fitted on tankers companies operated their ships and water washed their tanks with the tank atmosphere in either of these two conditions depending on the operator's preference. One of the disadvantages of operating in a too rich atmosphere, as represented by point A in Fig. 3, is that in the case of structural damage, such as in a collision, the hydrocarbon atmosphere within a tank could, in a sense, fall out of the punctured tank and gradually be replaced by air which could cause the tank atmosphere to increase in oxygen content and thereby pass into the flammable range. This, together with the fractured plating

possibly rubbing together to give a source of ignition in the form of sparks, could result in an explosive condition.

It will also be seen that provided the oxygen content is kept below, say, 11% the flammable atmospheres can be avoided. In practice and in order to provide a margin of safety the oxygen content of the inert gas being supplied to the cargo tanks is normally expected to be 5% by volume and an alarm is given if the oxygen content should reach 8%.

1.1.7 Source of Inert Gas

Now where can a gas having an oxygen content less than 11% be obtained on board ship? Fortunately there are various means of doing so, some of which will be described in more detail later on.

For the moment it is intended to concentrate on a flue gas system such as that shown diagrammatically in Fig. 4 and which has been annotated for compliance with Part 5, Chapter 15, Section 7 of the Rules.

To set the scene the following is a description of an inert gas system in general terms.

Firstly, oil and air in proportions of approximately one part oil to 16 parts air by weight are burned under automatic combustion control in the boiler to give an exhaust gas which contains approximately 5% oxygen. This hot gas is then cooled and washed in a scrubber and blown by means of blowers or fans through pipework and via various safety devices into the cargo tanks to replace the air and render the tank atmosphere inert.

Since the Rules for inert gas systems are very much self contained it is proposed to take advantage of this fact and quote each paragraph in italics followed by comments or interpretations and also to include for the guidance of surveyors a reference to Regulation 62 of Chapter II-2 of the 1981 Amendments to SOLAS 1974 as applicable to both NEW and EXISTING inert gas systems. It is hoped that this latter arrangement may be of particular use for those surveyors who are, from time to time, engaged in filling up the questionnaire required by the International Conventions Safety Equipment Department, viz ICD/ICL/124, dated 8th August, 1984, which has already been mentioned.

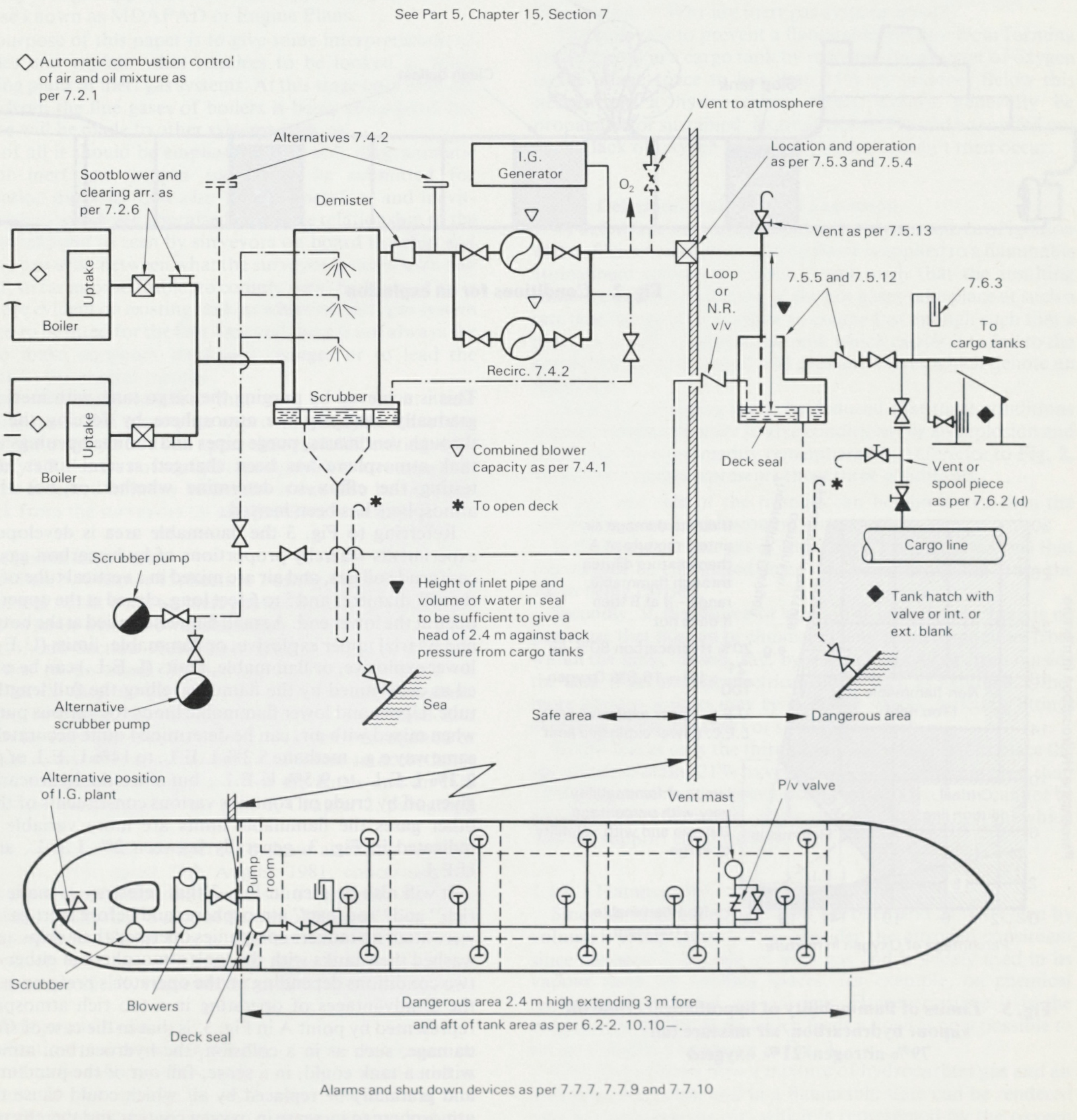


Fig. 4 Arrangement of an Inert Gas System (flue gas type)

1.1.8 Submission of Plans

To begin with it is first necessary to determine which plans have to be submitted and these are indicated in paragraph 5, 15, 1.1.2 of the Rules. These plans should be of diagrammatic type only except where the Rules require individual items of equipment to be approved such as a deck water seal or non-return valve.

These diagrams should, as far as possible, show the position of the equipment and pipework as it will be located on board the ship. This means that the boilers, scrubber, blowers and auto-closing bulkhead valve should be shown in logical sequence situated within the safe area and the deck seal and piping etc. to the cargo tanks shown within the dangerous area. This is stressed since in one case several years ago it was discovered, only accidentally, that the diagrammatic plans were not representative of the arrangements on board and as a result there were inherent dangers in the system which required rectification before approval could be given.

1.1.9 Society's Rules for Inert Gas Systems

PART 5, CHAPTER 15, SECTION 7
INERT GAS SYSTEMS FOR OIL TANKERS
Sub-Section 7.1 General

Paragraph 5, 15, 7.1.1. The following requirements apply where an inert gas system, based on flue gas, is fitted on board ships intended for the carriage of oil in bulk having a flash point not exceeding 60°C (closed cup test). Any proposal to use an inert gas other than flue gas, e.g. nitrogen, will be specially considered. (Regulation 62,—No equivalent)

This paragraph does not state that an inert gas system has to be fitted but only that where one is fitted it should comply with the Rules. Nevertheless, for fire protection purposes inert gas systems are now required to be fitted to oil tankers in accordance with Part 6, Chapter 4, Sub-Section 19.9 of the Rules. These Rules cover crude oil and product oil tankers and also combination carriers such as OBO and ORE/OIL. Additionally, an IGS notation may be required in respect of paragraphs 3, 2, 3.8.1 and 4, 9, 7.1.3 of the Rules.

Paragraph 5,15, 7.1.2. Ships complying with these requirements will be eligible for the additional notation "IGS" in the Register Book, see Pt 1, Ch 2. For inert gas systems on chemical tankers, see Section 8. (Regulation 62.—No equivalent).

Once again this paragraph does not state that the IGS notation will be assigned even if the Rules are complied with, but only that the ship will be "eligible" for the notation if desired. This implies that an owner does not have to have an IGS notation assigned if he does not wish it unless, of course, it is required for fire extinguishing purposes in which case Circular No 2381 should be used for guidance.

Paragraph 5, 15, 7.1.3. Throughout this Section the term "cargo tank" includes also "slop tanks". For definition of "Machinery spaces of Category 'A'", see Pt 6, Ch 4, 2.4.11. (Regulation 62 —A reference to the slop tank is made as a footnote at the bottom of the first page. For a definition of "Machinery spaces of Category 'A'", see the 1981 SOLAS Amendments Chapter II-1, Reg. 3(17) and Chapter II-2, Reg. 3(19).

This paragraph is self explanatory.

Sub-Section 7.2, Gas Supply

Paragraph 5, 15, 7.2.1. The inert gas may be treated flue gas from the main or auxiliary boiler(s), gas turbine(s), or from a separate inert gas generator. In all cases, automatic combustion control, capable of producing suitable inert gas under all service conditions, is to be fitted. (Regulation 62.4—It should be noted that there is no requirement for automatic combustion control).

Automatic combustion control is considered essential to ensure a steady state of inert gas supply without fluctuations in the amount of oxygen.

Since gas turbines and separate inert gas generators are mentioned Figures 5 and 6 have been included to show typical arrangements.

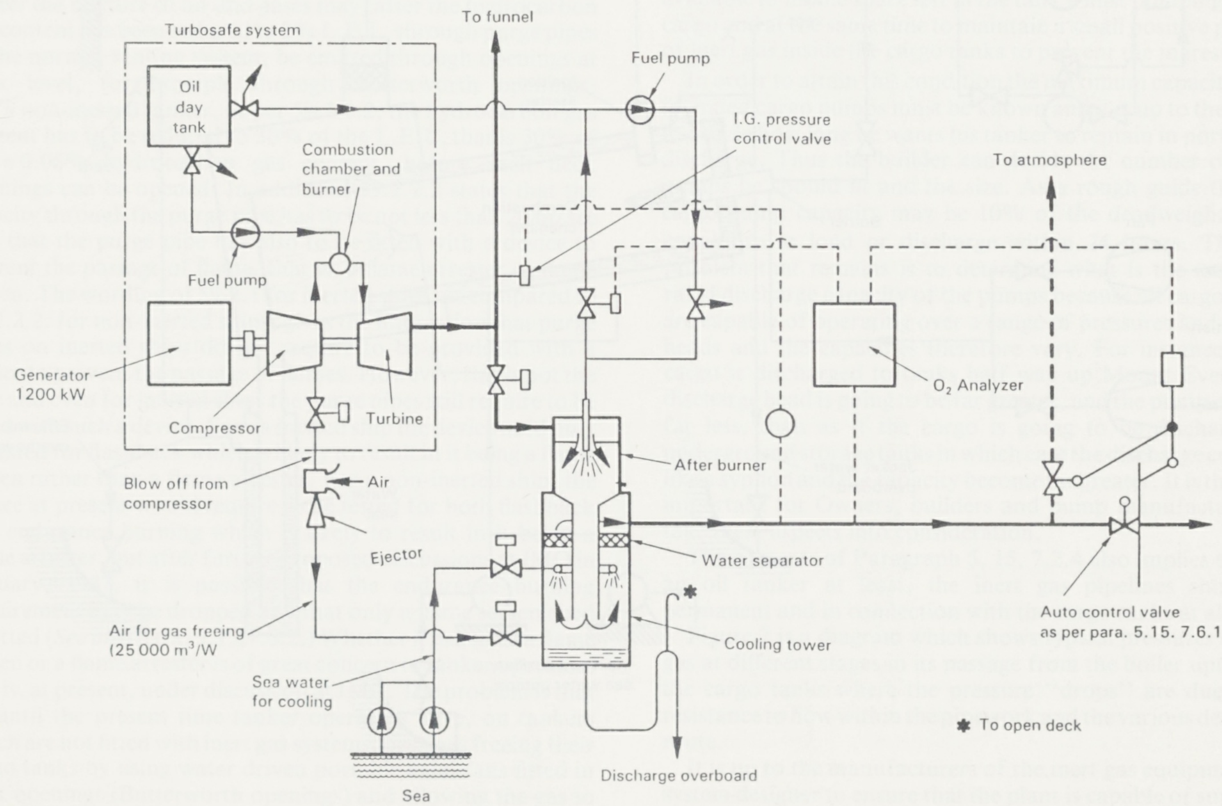


Fig. 5 Inert gas from a gas turbine

The gas turbine inert gas system utilises the clean exhaust from the gas turbine which has an oxygen content of approximately 16% by volume. Therefore it is necessary to fit an after-burner which reduces the oxygen content to 1 to 2%. As can be seen the after-burner is fitted within the scrubber or cooling tower. This system can also be adopted, by the addition of drying and refrigerating systems, to give a high quality inert gas for use on LNG/LPG liquefied gas carriers having a dew point of $-45°C$, an oxygen content of 0.5% and a capacity of up to 25,000 m^3/hours. Other advantages are that there are no moving parts, high reliability, easy operation, low maintenance costs, no electric power requirements and fresh air for gas freeing is taken from the gas turbine compressor.

An inert gas generator system, as distinct from a flue gas system is, as shown in Fig. 6, a self contained unit, utilising a burner encased within a scrubber, or cooling tower in a similar manner to that shown in the gas turbine system.

The exhaust gas is generally much cleaner than in a flue gas system since the oil is normally a distillate fuel which can be burned very efficiently under tight control to give complete combustion so that soot is virtually absent thereby avoiding the corrosive and erosive effects of sulphur and solid particles.

Inert gas generator systems are commonly fitted on board chemical tankers and liquefied gas carriers but also on crude oil tankers either as a large main unit and/or as a smaller topping up unit. The main unit would be used during unloading of the cargo oil when large volumes of inert gas are required to keep an overpressure of inert gas in the cargo tanks or during other operations such as tank washing. The topping up unit would be used, perhaps in association with a flue gas system, during either the loaded or ballast passage to maintain an inert atmosphere within the cargo tanks without, for example, having to start up the large system for what will probably be a very short period of time. Furthermore, if it were proposed to use a flue gas system for topping up purposes at sea when, for example, the steam driven cargo pumps are not in use and the boiler is therefore on light load, it is unlikely that the required quality of exhaust/inert gas having low oxygen content could be obtained.

Until recently the exhaust gas from main diesel engines was considered unsuitable for use in inert gas systems due both to the uneveness of the flow and the high oxygen content of the exhaust gas, about 12%. However, in a similar manner to the gas turbine system, at least one firm has now designed an inert gas generator capable of after-burning the diesel engine exhaust gas with a resultant saving in the generator oil fuel costs as a result of having only to reduce the oxygen content from 12% to say 5% instead of from 21% in air, to 5%.

Paragraph 5, 15, 7.2.2. Two fuel oil pumps are to be fitted to the inert gas generator. One fuel pump only may be accepted provided sufficient spares for the fuel oil pump and its prime mover are carried on board to enable any failure of the fuel oil pump and its prime mover to be rectified by the ship's crew. (Regulation 62.7.2).

This paragraph relates to the fuel pump(s) of an inert gas generator and not to the fuel pumps of a main boiler. The extent of spares is usually a matter for agreement between the manufacturer and the shipbuilder or owner but would normally consist of a spare motor and pump and burner nozzle assembly.

Paragraph 5, 15, 7.2.3. The inert gas system is to be capable of:

(a) *inerting empty cargo tanks by reducing the oxygen content of the atmosphere in each tank to a level at which combustion cannot be supported;* (Regulation 62.2).

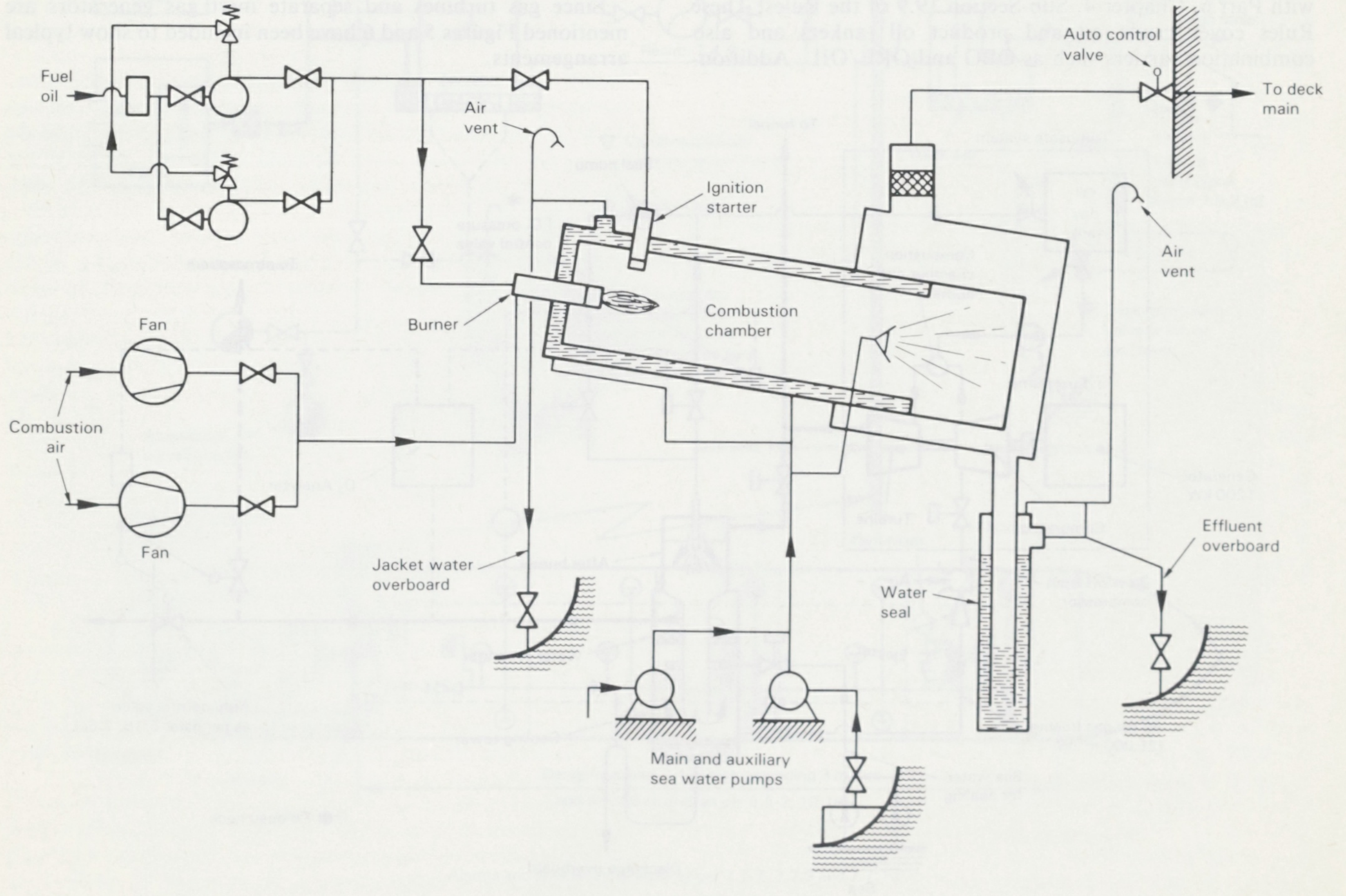


Fig. 6 Inert gas from an inert gas generator

As previously mentioned the oxygen content is reduced by replacing the atmosphere in the tanks with an inert gas having an oxygen content of approximately 5%.

(b) maintaining the atmosphere in any part of any cargo tank with an oxygen content not exceeding 8 per cent by volume and at a positive pressure at all times in port and at sea except when it is necessary for such a tank to be gas free.

This could involve testing the atmosphere in each tank for oxygen content. There is an alarm condition at 8% oxygen in the inert gas being supplied to the tank (*See* 5, 15, 7.7.7(e)), but not at the tanks themselves and although gas having 5% oxygen content is being supplied it is still possible to have 8% oxygen content within the cargo tanks due to leakage of air into the system en route. The "positive pressure at all times" is ensured by paragraph 5, 15, 7.7.4.

(c) eliminating the need for air to enter a tank during normal operations except when it is necessary for such a tank to be gas free;

This is effected by maintaining an overpressure of inert gas in the cargo tanks at all times and by ensuring that the low pressure alarms referred to in paragraphs 5, 15, 7.7.7(h) and 5, 15, 7.7.14 are operational.

(d) purging empty cargo tanks of hydrocarbon gas, so that subsequent gas freeing will at no time create a flammable atmosphere within the tank.

This means that any hydrocarbon gases should be purged with inert gas until the concentration of hydrocarbon vapours being emitted through the purge pipes or venting system has been reduced to 2% by volume. At this stage the efflux gases should no longer be flammable and air can be introduced to gas free the tanks for entry.

Attention is also drawn to regulation II-2/59.2 of the 1981 Amendments to SOLAS which makes the distinction between gas freeing an inerted and a non-inerted tanker. For an inerted tanker the mixture of air and gases may, after the hydrocarbon gas content has been reduced to 2% L.E.L. through purge pipes or the normal venting system, be emitted through openings at deck level, for example through Butterworth openings. For a non-inerted tanker, as per 59.2.2.2, the hydrocarbon gas content has to be reduced to 30% of the L.E.L, that is 30% of 2% = 0.06% hydrocarbon gas content, before such deck openings can be opened. In addition, 59.2.2.2 states that the velocity through the purge pipe has to be not less than 20 m/sec and that the purge pipe has also to be fitted with a device to prevent the passage of flame, that is, a flame arrester or flame screen. The wording of 59.2.1 for inerted ships, as compared to 59.2.2.2. for non-inerted ships, gives the impression that purge pipes on inerted ships do not require to be provided with a device to prevent the passage of flames. However, this is not the case and even for inerted ships the purge pipes still require to be fitted with such a device. For an inerted ship the device need only be tested for flashback which is likely to result in it being a flame screen rather than a flame arrester. For a non-inerted ship, the device at present would require to be tested for both flashback and endurance burning which is likely to result in it being a flame arrester, but after further proposed discussions at IMO in January, 1987, it is possible that the endurance burning requirement may be dropped and that only a flame screen need be fitted (*See also* para. 5,15,7.6.2.) Whether it has to be a flame screen or a flame arrester is of great concern to tanker operators and is, at present, under discussion at IMO. The problem is that up until the present time tanker operators have, on tankers which are not fitted with inert gas systems, been gas freeing their cargo tanks by using water driven portable axial fans fitted in deck openings (Butterworth openings) and allowing the gas to escape through open hatch covers. By this method the gas freeing air is able to penetrate and circulate through the depth of the cargo tanks and supply the large volume of air required to efficiently gas free. However, this method is no longer considered to be within the intent of Regulation 59.2.2.2. and henceforth such gas freeing will have to be through either purge pipes fitted with flame screens or flame arresters or through a main venting system having fixed centrifugal fans. The flame screens or arresters fitted to the purge pipes create a great resistance to flow and it has been found in practice that it has not been possible to efficiently gas free using portable fans since the air stream tends to short-circuit across the top of the tank. In addition, paragraphs 1.2.5.3 and 2.5.3.2 of MSC/CIRC.373 indicate that flame screens have to be fitted even over the inlets of portable or fixed fans increasing the resistance to gas freeing even further. It seems therefore that, unless the design of portable fans can be changed to operate efficiently against these resistances, the only alternative will be to use the larger capacity and more powerful centrifugal fans. Unfortunately, since there are no known portable centrifugal fans at the moment this would mean a fixed centrifugal fan, or fans, connected to a fixed deck mounted venting system. The tanker fitted with an inert gas system has no problem in this respect since the blowers incorporated into the inert gas system can be used for gas freeing. The biggest problem would be with chemical tankers which already have a great deal of pipework on their decks and it remains to be seen whether IMO will relax the requirements for fitting flame screens or flame arresters, on inlets and outlets to purging arangements having regard, for example, to the fact that tanks are not purged and/or gas freed every day and that, for most of a tanker's working life, such purging openings remain closed. By comparison, venting devices such as pressure/vacuum valves etc, have a potential for being open at any time and therefore require greater protection.

Paragraph 5,15, 7.2.4. The system is to be capable of delivering inert gas to the cargo tanks at a rate of at least 125 per cent of the maximum rate of discharge capacity of the ship expressed volume. (Regulation 62.3.1.).

The intention here is to ensure that sufficient inert gas is available to fill the space left in the tank whilst pumping out the cargo and at the same time to maintain a small positive pressure of inert gas inside the cargo tanks to prevent the ingress of air.

In order to attain this condition the maximum capacity of the intended cargo pumps must be known and it is up to the Owner to decide how long he wants his tanker to remain in port during discharge. Thus the builder can decide the number of cargo pumps he should fit and the size. As a rough guide the total cargo pump capacity may be 10% of the deadweight or the capability to load or discharge within 24 hours. The only problem that remains is to determine what is the maximum rated discharge capacity of the pumps because all cargo pumps are capable of operating over a range of pressures and suction heads and the capacities therefore vary. For instance, if the cargo is discharged to tanks half way up Mount Everest the discharge head is going to be far greater, and the pump capacity far less, than as if the cargo is going to be discharged to underground storage tanks in which case the discharge could act like a syphon and the capacity become far greater. It is therefore important for Owners, builders and pump manufacturers to take these aspects into consideration.

The contents of Paragraph 5, 15, 7.2.4 also implies that, on an oil tanker at least, the inert gas pipelines should be permanent and in connection with the cargo tanks at all times.

Figure 7 is a diagram which shows typical pressures of inert gas at different stages in its passage from the boiler uptakes to the cargo tanks where the pressure "drops" are due to the resistance to flow within the pipework and the various devices en route.

It is up to the manufacturers of the inert gas equipment and system designer to ensure that the plant is capable of supplying the required quantity of gas (*see also* comments on 5, 15, 7.7.6 (g)).

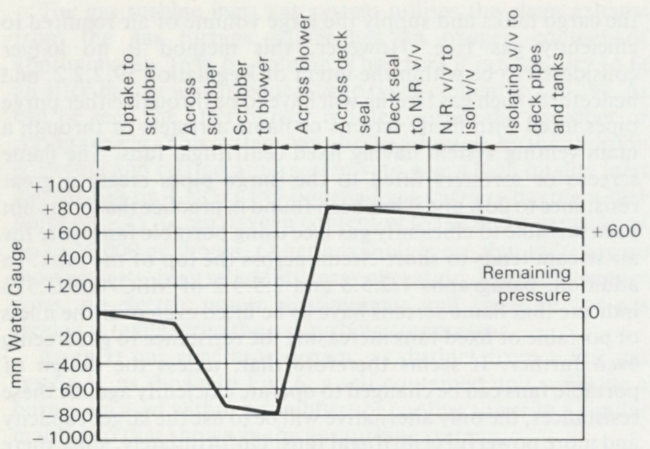


Fig. 7 Inert gas pressure diagram

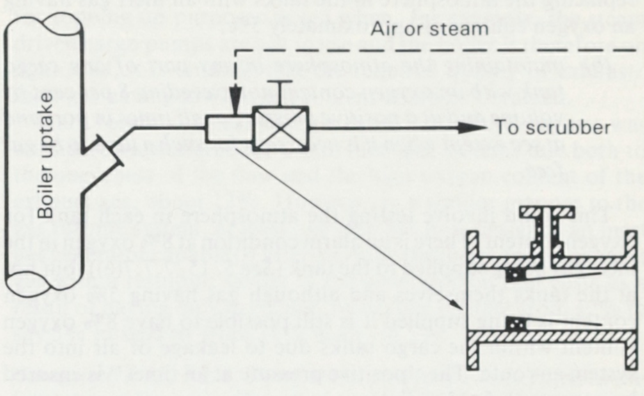


Fig. 8 Valve seat cleaning

Paragraph 5, 15, 7.2.5. The system is to be capable of delivering inert gas with an oxygen content of not more than 5 per cent by volume in the inert gas supply main to the cargo tanks at any required rate of flow. (Regulation 62.3.2.—Need not be complied with for existing I.G. systems. *See* 62.20.1 and 62.20.2).

Having automatic combustion control as referred to in paragraph 5, 15, 7.2.1 would normally ensure that the amount of oxygen in the boiler exhaust gases would be about 4% to 5%. However, should the oxygen content rise due to control malfunction it will be necessary to get rid of that over-oxygenated portion of inert gas. For this reason a connection such as that shown just before the bulkhead valves in Fig. 4 is usually fitted to enable the gas to be discharged to atmosphere, most probably back to the funnel. Should no such connection be fitted then it would be possible to open the bulkhead valve and allow this portion of inert gas to be vented to atmosphere through the venting system. Re-circulating this portion of inert gas back to the scrubber via the re-circulating line, also shown in Fig. 4, would not materially reduce the oxygen content and, indeed, this line is used for another purpose as explained under paragraph 5, 15, 7.4.2.

Paragraph 5, 15, 7.2.6. Flue gas isolating valves are to be fitted in the inert gas supply mains between the boiler uptakes and the flue gas scrubber. These valves are to be provided with indicators to show whether they are open or shut, and precautions are to be taken to maintain them gas tight and keep seatings clear of soot. Arrangements are to be made to ensure that boiler soot blowers cannot be operated when corresponding flue gas valve is open. (Regulation 62.5).

Various proposals have been seen for keeping the uptake valve seatings clear of soot but the two most common arrangements are:

(i) an annular ring within the pipe adjacent to the valve seat through which either air or steam can be blown—*see* Fig. 8 or

(ii) a manhole next to the valve to enable manual cleaning to take place when the plant is shut down. (*See* Fig. 4)

Bellows pieces are often fitted between the boiler uptake valve and the scrubber and since the temperatures will be in the region of 315°C (600°F) these will be of metallic construction. However, since the scrubber cools the hot gases to within 5°C of the sea water temperature any bellows pieces fitted after the scrubber could be made of non-metallic material such as neoprene protected against fire by metallic shielding.

With regard to preventing the sootblowers being operated when the corresponding uptake valve is open, this is to ensure that additional soot displaced during the sootblowing operation and containing corrosive and erosive products will not be drawn into the scrubber and thereafter into the inert gas system where it would cause great damage. Until the 1985 revision to the Society's Rules were published this was allowed to be controlled by a suitably worded notice board located adjacent to the sootblower master steam valve. However, since the inclusion of the wording of IMO Regulation 62 the intention would appear to require an electric/mechanical interlock to "ensure" that the sootblower steam valve and the corresponding uptake valve cannot be open at the same time. There are boilers, usually auxiliary boilers, which do not have sootblowers fitted and the soot is washed away by means of a hose when the I. G. plant is shut down.

Sub-Section 7.3 GAS SCRUBBER

Paragraph 5, 15, 7.3.1. A flue gas scrubber is to be fitted which will effectively cool the volume of gas specified in 7.2.4. and remove solids and sulphur combustion products. The cooling water arrangements are to be such that an adequate supply of water will always be available without interfering with any essential services on the ship. Provision is also to be made for alternative supply of cooling water. (Regulation 62.6.1.).

There are various makes and types of scrubbers three of which are shown in Figures 9(a), (b), and (c). The choice of type depends on the type of tanker, the cargoes to be carried, the combustion equipment and the claims of the various manufacturers for the greater efficiency of their equipment compared to others. In any event most scrubbers, as designed, can be depended upon to wash out 90 to 99% of the sulphurous (corrosive) and solid (erosive) products of combustion by the action of the sea water and variations in speed and changes of direction of the gas during its passage through the scrubber.

Of the three types shown it is claimed that the bubble cap and agglomerating types have the highest efficiency in reducing the sulphur and solids content but this is at the expense of a high pressure drop which indicates a high resistance to flow. Nevertheless the high cleanliness could, for instance, be important when carrying sophisticated products. On the other hand the packed scrubber is claimed to be simpler, more reliable and has a lower pressure drop. This could, however, cause greater carry over of the water content.

The removal of the sulphur content in the gas seems to depend on the water rate whereas the removal of the solids is more affected by the pressure drop across the scrubber where high resistance to gas flow would give time for the solids to precipitate out. As a general figure the amount of solids in the flue gas or "solid burden" as it may be called, will probably be in the region of 150 milligrams per cubic metre of gas with efficient burning and reliable combustion control. However,

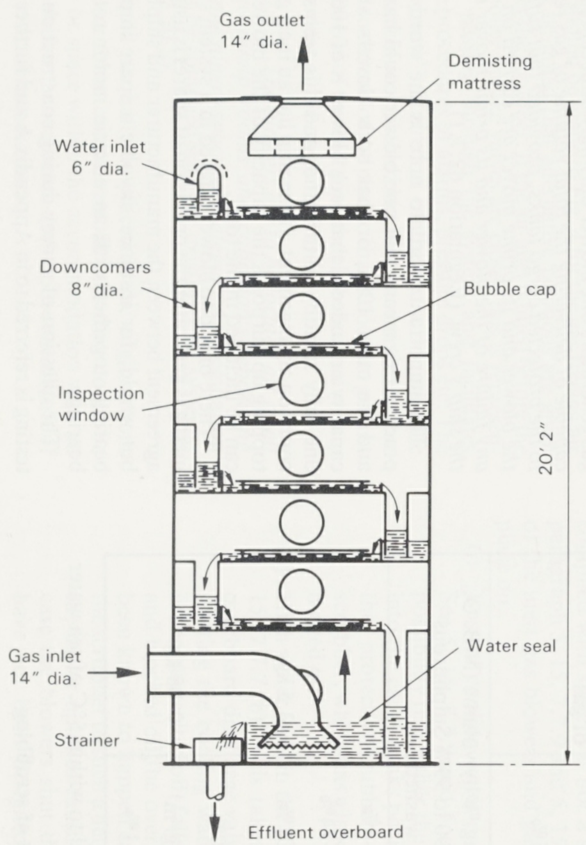


Fig. 9(a) Bubble cap type scrubber

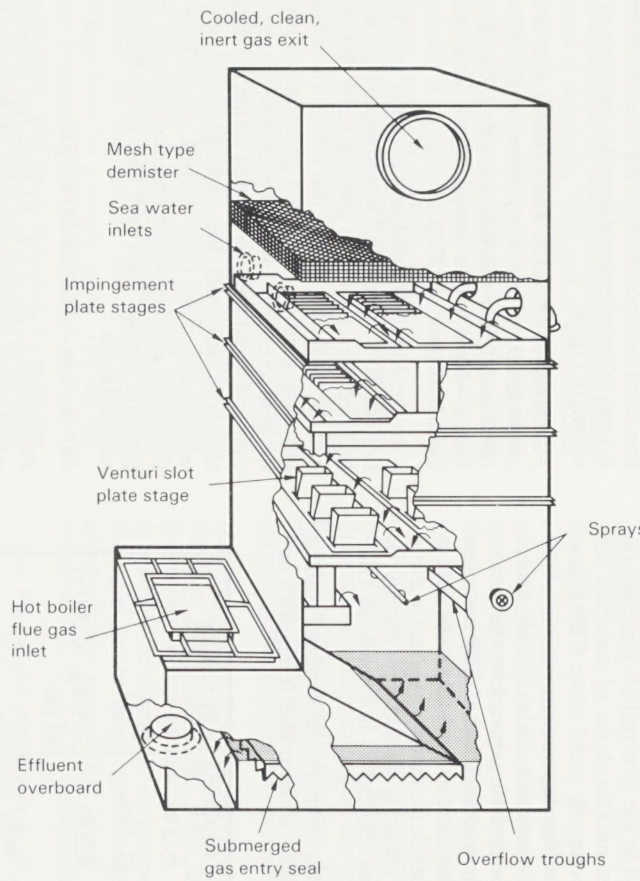


Fig. 9(b) Impingement and agglomerating scrubber

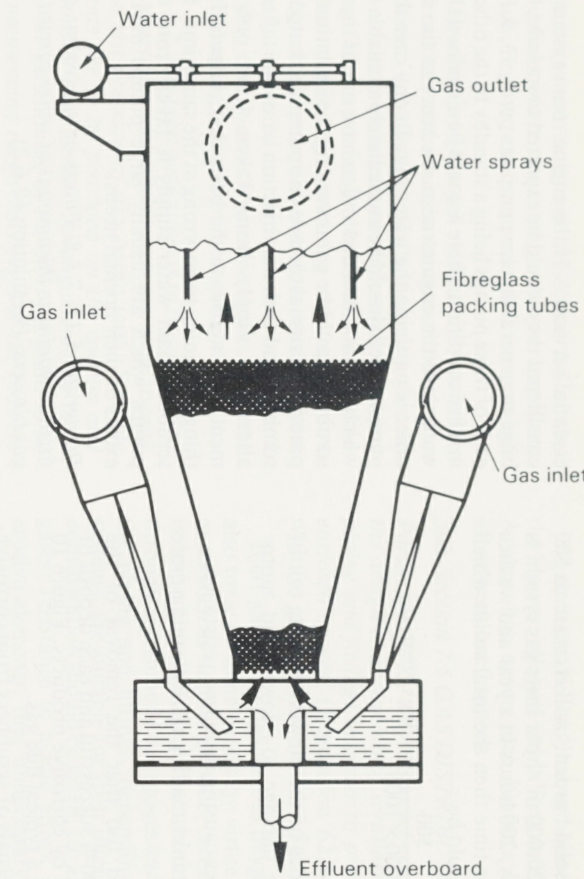


Fig. 9(c) Packed spray type scrubber

after some time in use the "solid burden" will increase to 500 mg/m^3 of gas, or more. If a 20,000 m^3/hour inert gas system is considered to operate for, say, 200 hours in a year and washes out 98% of the solids in that time then the total solids which could be left in the system would be

$$\frac{2}{100} 20{,}000 \times 200 \times \frac{500}{1000 \times 1000} = 40\text{kgs/year.}$$

These solids could then:

(i) Adhere to valve seats.

(ii) Deposit upon the blades of the blowers causing out of balance and possible damage to blower and motor bearings.

(iii) Deposit in the pipelines causing increased resistance.

(iv) Deposit in the cargo tanks making tank cleaning more difficult.

It should be remembered too that the removal of the sulphurous products is not 100% successful and each deposit of solids could be accompanied by corrosive products. Figure 10 shows the effect of scrubbing a typical flue gas.

In order to protect the internals of scrubbers from corrosive and erosive attack it is normal practice to coat any mild steel with neoprene or plastics and to use incorrodible materials on other critical parts. The most suitable choice of materials has been an evolutionary one over many years. Since GRP (Glass Reinforced Plastics) exhibits a high degree of resistance to both corrosive and erosive attack a scrubber made of GRP, except for the bottom section, was accepted in principle by the Society about ten years ago. The only drawback is that GRP has a low melting point and it would therefore be necessary to be able to stop the water supply from outside the machinery space to avoid flooding or other damage occuring should fire destroy or damage the scrubber housing. So far as is known no GRP scrubber has yet been fitted.

Paragraph 5, 15, 7.3.1, also refers to the scrubber cooling water supply and to the words "without interfering with any essential services". If the engine room systems in a tanker are considered they would be expected to include, for instance, two bilge or general service pumps as per 5, 13, 6.1.1. of the Rules, each of these pumps being a standby to the other except perhaps in a fire condition where it could be visualised that one is putting water into the engine room via hoses at the same time as the other is pumping the water out. In any case these are essential pumps which should be immediately available for the duties for which they are intended and to have one of them operating on a scrubber would be contravening the intention of the Rules. In practice it is usual to have one pump dedicated to supplying the scrubber and then no objection would be taken to the required alternative supply being taken from one of the pumps mentioned above on the basis that, hopefully, not too many things are going to go wrong at the same time. In some cases the scrubber main water supply is taken from foam or salvage pumps which are usually large pumps having capacities in excess of their normal needs.

Composition before scrubbing % by volume		
Oxygen	2–4%	
Carbon Dioxide	12–14%	
Sulphur Dioxide/ Trioxide	0,20%	= 35kg/10,000 m^3 of gas
Nitrogen	80%	
Solids	300mg/m^3	= 3kg/10,000 m^3 of gas
Water vapour	5%	
Composition after scrubbing % by volume (N.B. O_2, CO_2 and N unchanged but 90 to 99% Sulphurous content, and most of solids washed out.		
Oxygen	2–4%	
Carbon Dioxide	12–14%	
Sulphur Dioxide/ Trioxide	0,02%	= 3.5 kg
Nitrogen	80%	
Solids	8mg/m^3	= 0·08 kg
Water vapour	0,1%	

Temperature of gases reduced to within 5°C of sea water

Fig. 10 Effect of scrubbing

Paragraph 5,15, 7.3.2. Filters or equivalent devices are to be fitted to minimize the amount of water carried over to the inert gas blowers. (Regulation 62.6.2).

The filters or equivalent devices, may be either mesh type demisters incorporated into the scrubber as shown in Figures 9 (a) and (b) or an external demister, possibly of a vortex type, as shown in Fig. 4.

Paragraph 5, 15, 7.3.3. The scrubber is to be located aft of all cargo tanks, cargo pump rooms and cofferdams separating these spaces from machinery spaces of Category A. (Regulation 62.6.3—Need not be complied with for existing I.G. systems. *See* 62.20.1 and 62.20.2).

In the Author's opinion it is rather surprising that Regulations 62.20.1 and 62.20.2 would apparently allow a scrubber to be fitted forward of the automatically closing bulkhead valve and the reasoning behind this relaxation is not known. It is, in any case, considered to be a potentially dangerous relaxation because scrubbers have to be inspected internally from time to time and should one be located within the dangerous area over the cargo tanks and then opened up for inspection, the gases, which are assumed to be on deck, would have a direct route to the boiler uptakes where obviously sources of ignition are in plentiful supply. Each such case would have to receive special consideration.

Sub-Section 7.4 GAS BLOWERS

Paragraph 5, 15, 7.4.1. At least two blowers are to be fitted which together are to be capable of delivering to the cargo tanks at least the volume of gas required by 7.2.4. In the system with gas generators one blower only may be accepted if that system is capable of delivering the total volume of gas required by 7.2.4. to the protected cargo tanks, provided that sufficient spares for the blower and its prime mover are carried on board to enable any failure of the blower and its prime mover to be rectified by the ship's crew. (Regulation 62.7.1).

The important word to note is the word "together". In practice this means that one blower could have 90% capacity and the other 10%, or that both blowers are each of 50% capacity or, indeed, that each blower is of 100% capacity, the one fully standby to the other and this seems to be the most common arrangement. If the ship has an inert gas generator for topping up purposes the capacity of the blower(s) for this unit can be included in the total.

Where only one blower is fitted in a system supplied from an inert gas generator the provision of spares is usually a matter for agreement between the manufacturer and shipbuilder or owner but would, for instance, involve a spare impeller, shaft and bearings together with an electric motor rotor, bushes and bearings.

The question of survey during construction of blowers and testing is referred to in Appendix A and further guidance on the

materials of construction can be found in Section 3.5 of the IMO Inert Gas Guidelines where, for example, blower impellers made of aluminium bronze are recommended to combat erosion.

Blowers are the heart of an inert gas system and, as in the human frame, suffer a lot of abuse due, in the case of blowers, to corrosion and erosion; unbalancing of the impeller due to deposits and water entrainment in carry over from the scrubber resulting in bearing failures; damage due to structure borne vibration both when the motor is running or stopped and inadequate drainage. It is essential therefore that blowers should be regularly inspected to ensure their continued efficiency under trying conditions. When one thinks of it, the blowers are the only moving parts of an inert gas system.

Paragraph 5, 15, 7.4.2. The inert gas system is to be so designed that the maximum pressure which it can exert on any cargo tank will not exceed the test pressure of any cargo tank. Suitable shut-off arangements are to be provided on the suction and discharge connections of each blower. Arrangements are to be provided to enable the functioning of the inert gas plant to be stabilised before commencing cargo discharge. If the blowers are to be used for gas freeing, their air inlets are to be provided with blanking arrangements. (Regulation 62.73.).

The design of equipment in an inert gas system is in many instances, as indicated in Appendix "A", not of classification concern and, for example, it is the manufacturer's responsibility to design and supply blowers which will not exert too great a pressure on any cargo tank. As can be seen from Fig. 7 it is unlikely that the pressure of inert gas at entry into the cargo tanks will ever approach 0.24 kgf/cm^2 (3.5 p.s.i) which is the pressure to which cargo tanks are normally tested.

The "shut-off arrangements" referred to are usually butterfly valves of metallic or resilient seated type since they are the most convenient to use but sometimes spectacle flanges are utilised instead.

Regarding the word "stabilised", reference to Fig. 4 shows a re-circulating line which returns from the blower discharge line to the scrubber. This line is fitted to enable the plant to be stabilised before opening up the bulkhead valve and refers particularly to blowers which are driven by steam turbines instead of electric motors. Unlike electric motors steam turbines cannot just be switched on and require time to be warmed through before settling down to a steady speed. During this time, and to prevent overheating of the fans, the inert gas is re-circulated to the scrubber. This line is also used, for example, when the full output of the blowers is not required or to allow time, perhaps up to 2 hours, for the oxygen analyser to obtain an accurate reading before opening the bulkhead valve.

Again reference should be made to Fig. 4 for the gas freeing air inlets where it will be noted that alternative positions are given either before or after the scrubber. If the air inlet is fitted before the scrubber and the boiler uptake valves are closed this is representative of flue gas passing through the system and facilitates preliminary testing procedures of alarms, pressure drops etc. The "blanking arrangements" referred to can be just a thin metallic blank since no pressures are involved but it is recommended that the blank be hinged to the end of the pipe so that it does not get lost or mislaid. The open end of the inlet pipe should be located within a safe area on deck clear of the cargo tank area.

Paragraph 5, 15, 7.4.3. The blowers are to be located aft of all cargo tanks, cargo pump rooms and cofferdams separating these spaces from machinery spaces of Category A. (Regulation 62.7.4.—Need not be complied with for existing I.G. systems. *See* 62.20.1 and 62.20.2).

The blowers would normally be placed in the engine room together with the scrubber but on retrofit jobs they could be located in a room or space within the accommodation with an entrance from the deck or in a separate deck house usually aft of or at the side of the accommodation.

Again it can be said that as regards the relaxation in locating the blowers allowed for in Regulations 62.20.1 and 62.20.2 the Author has never seen blowers located in other than the aforementioned spaces.

Sub-Section 7.5 GAS DISTRIBUTION LINES

Paragraph 5, 15, 7.5.1. Special consideration is to be given to the design and location of scrubber and blowers with relevant piping and fittings in order to prevent flue gas leakages into enclosed spaces. (Regulation 62.8.1—Need not be complied with for existing I.G. systems. *See* 62.20.1).

Inert gas, being deficient in oxygen, is an asphyxiant and is also toxic to some extent by virtue of some of its constituent gases such as sulphur dioxide, carbon monoxide or nitrogen. It is important therefore that the inert gas lines do not pass through accommodation spaces and that the materials used for the piping and fittings, depending on their location in the systems, are suitable for their intended service. In this respect consideration should be given, for example, to resistance to hot gases; resistance to acidic attack or other corrosive and/or erosive elements; that unnecessary bends or branches where damp acidic soot could accumulate are avoided and that pipework should be of heavy gauge steel pipe suitably coated internally.

Paragraph 5, 15, 7.5.2. To permit safe maintenance, an additional water seal or other effective means of preventing flue gas leakage is to be fitted between the flue gas isolating valves and scrubber or incorporated in the gas entry to the scrubber. Regulation 62.8.2—Need not be complied with for existing I.G. systems. *See* 62.20.1).

In addition to the "effective means" quoted in the paragraph, removing a section of pipe between the boiler uptake valves and the scrubber and blanking off the open ends of fitting spectacle flanges could be equally effective. The important point is to test the atmosphere inside the scrubber to ensure 21% oxygen content before entry by any personnel.

Paragraph 5, 15, 7.5.3. A gas regulating valve is to be fitted in the inert gas supply main. This valve is to be automatically controlled to close as required in 5, 15, 7.7.9 and 5, 15, 7.7.10. It is also to be capable of automatically regulating the flow of inert gas to the cargo tanks unless means are provided to automatically control the speed of the inert gas blowers required in 5, 15, 7.4.1. (Regulation 62.9.1).

See Fig. 4 and paragraph 5, 15, 7.5.4 for location of the automatically regulating/closing valve. For existing inert gas systems it has been agreed that the regulating aspect need not be complied with since it is not a safety feature. As indicated in paragraph 5, 15, 7.7.9 and 5, 15, 7.7.10, automatic shut down of the inert gas blowers and gas regulating valve should take place on:

(i) Loss of cooling water pressure or flow to the scrubber. See 5, 15, 7.7.9 and 5, 15, 7.7.7. (a). This covers the possibility that either the uncooled hot gases could pass into the cargo tanks posing a source of ignition or that the protective rubber or plastic lining inside the scrubber, which has a low melting temperature, could be melted off.

(ii) High water level in the scrubber. See 5, 15, 7.7.9 and 5, 15, 7.7.7.(b). This takes into account either that the overboard discharge valve has not been opened before starting the cooling water pump or that the scrubber lining has melted off, fallen to the bottom of the scrubber and blocked off the overboard discharge line. This has been known to happen causing the water to build up in the scrubber to such a height that it overflowed back into the boiler uptakes causing extensive damage. In either case the blowers shut down automatically since they have not been designed for pumping water and the

regulating valve closes to prevent the possibility of large amounts of water passing into the cargo tanks where, especially in the case of product tankers, it could contaminate the cargo.

(iii) High gas temperature on the discharge side of the blowers. *See* 5, 15, 7.7.9 and 5, 15, 7.7.7.(c). This takes into account that either the supply of cooling water has failed and that the resulting high temperature could be a source of ignition in the presence of hydrocarbon gases or that heat sensitive materials, which may have been used, are no longer protected. The alarm condition is usually set to about 65°C and the shutdown condition to about 75°C.

(iv) Failure of the inert gas blowers. *See* 5, 15, 7.7.10 and 5, 15, 7.7.7(d). This takes into account that, since the pressure of the inert gas supply will have dropped, the hydrocarbon gases in the cargo tanks may flow back through the inert gas lines and thereby pass from a dangerous space into a safe one.

Paragraph 5, 15, 7.5.4. The valve referred to in 7.5.3 is to be located at the forward bulkhead of the forwardmost gas safe space through which the inert gas supply main passes. (Regulation 62.9.2.—Need not be complied with for an existing I.G. system. *See* 62.20.1).

The automatically controlled valve referred to in the previous paragraph is important because, by being placed on the bulkhead of the forwardmost gas safe space through which the inert gas line passes, it forms the positive means of closure between the safe and dangerous areas on a tanker.

In many cases, especially on existing tankers where an inert gas system is being fitted for the first time (being retrofitted) it is sometimes difficult to arrange for the gas regulating valve to be fitted at the bulkhead and a length of pipe between the valve and the bulkhead has been accepted provided the pipe is of one length and of substantial thickness, say 8 to 10mm thick. The reason for the single length is to limit the number of joints within the safe area which form potential points of leakage should there be a back flow of hydrocarbon gases from the cargo tanks.

The location of the valve at the bulkhead of the "forwardmost safe space" referred to takes into account the forward bulkhead of the engine room when the inert gas plant is located in the engine room but also takes into account the circumstance where the inert gas plant has been located in another space due to lack of room in the engine room. In this case the valve is fitted at the bulkhead of that space. See Fig. 4 for an example of Alternative Position of I.G. Plant.

Paragraph 5, 15, 7.5.5. At least two non-return devices, one of which is to be a water seal, are to be fitted in the inert gas supply main, in order to prevent the return of hydrocarbon vapour to the machinery space uptakes or to any gas safe spaces under all normal conditions of trim, list and motion of the ship. They are to be located between the automatic valve required by 7.5.3 and the aftermost connection to any cargo tank or cargo pipeline. (Regulation 62.10.1).

See Fig. 4. The reasons for having one water deck seal and one other non-return device, which is usually a metallic seating non-return flap valve, is that the non-return valve is considered to be effective against a back flow of cargo oil from the tanks and the water seal effective against a backflow of hydrocarbon gases from the cargo tanks. Since loading a tanker is a controlled operation, liquid overflows should not occur very often whereas the hydrocarbon gases can return as far as the deck seal, if all the necessary valves are not closed, every time the inert gas system is stopped. The continued efficiency of the deck seal is the more important of the two devices and this is given credence by the fact that when the Rules first came out it was allowed to have two non-return valves but this was later changed to the present requirements. It has also been known for the mechanical non-return valve to be damaged by corrosion which would allow hydrocarbon gases to pass back.

The deck seal, being a water seal, offers resistance to the passage of gases and especially to the passage of air when gas freeing since this resistance creates a pressure drop and pressure drops mean less gas or air volume getting to the tanks. Therefore deck seals have been developed which drop, suck or expel the water out of the seal while the inert gas is flowing and automatically re-establish the seal either when the flow stops or there is a significant drop in pressure such as when a blower slows down.

Figures 11(a), (b) and (c) show three different types of deck seal one of which is a dry type, one a semi-dry type and one a wet type.

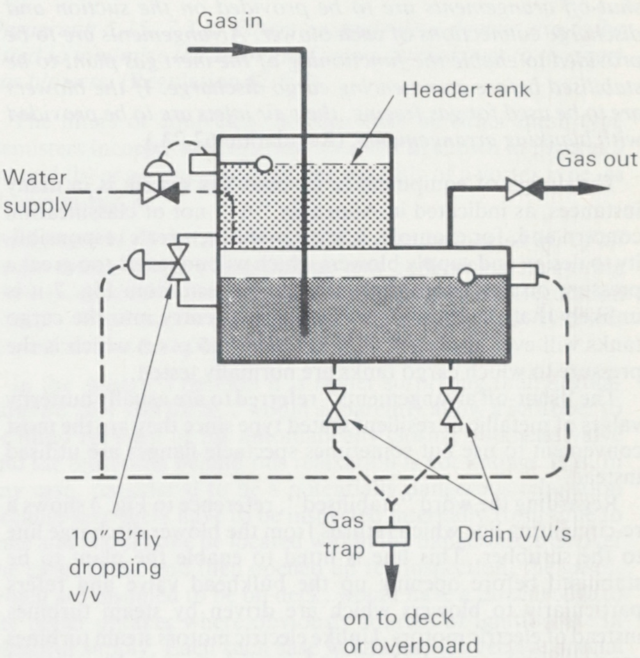


Fig. 11(a) Dry type deck seal

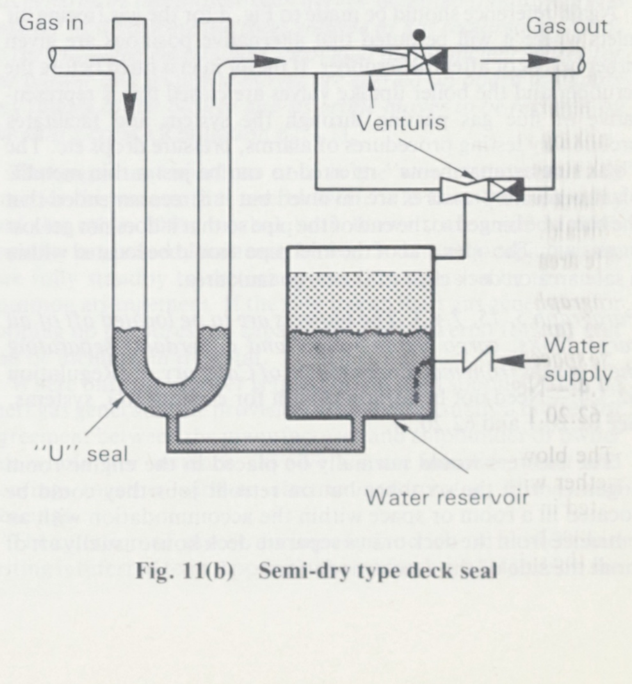


Fig. 11(b) Semi-dry type deck seal

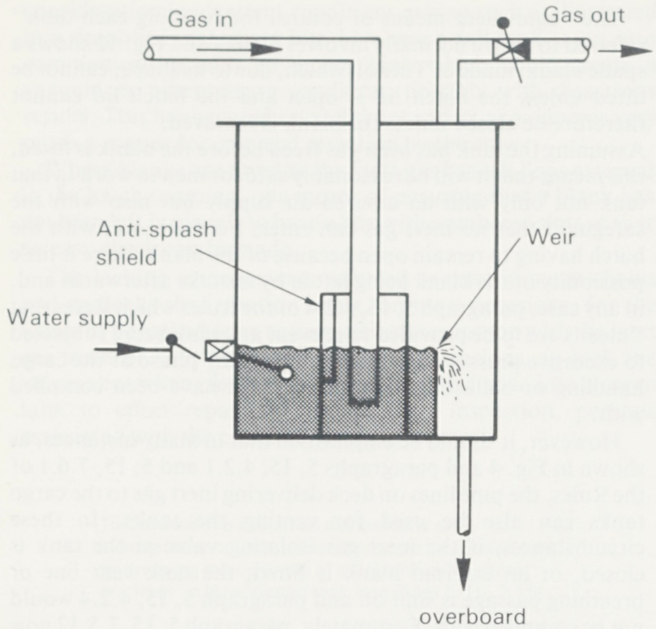


Fig. 11(c) Wet type deck seal

Regarding Fig. 11(a) the drain valves are connected to the blower starter such that the drain valves open and drain the seal when the blower starts and close again when the blower stops, at which time the water in the header tank will also automatically drop and re-establish the seal. The gas trap prevents the inert gas blowing out while the seal is in operation.

In Fig. 11(b) the inert gas bubbles through the water in the "U" bend and first passes through the bottom venturi which is a small and highly efficient one. This causes a vacuum condition in the water reservoir which allows the water in the "U" bend to be drawn into the reservoir after which the capacity and pressure of the inert gas now passing through the dry "U" bend is sufficient to open the large non-return valve in the main line.

Any appreciable drop in the pressure of the inert gas supply will decrease the vacuum condition in the water reservoir and allow the seal to be re-established.

The wet type of deck seal shown in Fig. 11(c) allows carry over of water to the deck lines and cargo tanks. This loss is automatically made up through a float level control valve and, on ordinary cargo oil tankers, the carry over of water to the deck lines and tanks is acceptable. However, it may be well to mention that drains should be provided at suitable positions in the deck lines to release any trapped water, for instance, at the port and starboard sides if the athwartship pipes follow the deck camber. Drains should, in fact, be fitted to any deck lines where it is anticipated that water or oil could accumulate.

Paragraph 5, 15, 7.5.6. The devices referred to in 7.5.5 are to be located in the cargo tank area on deck. (Regulation 62.10.2—Need not be complied with for existing I.G. Systems. *See* 62.20.1).

If the wording of this paragraph is read in conjunction with the wording of the previous paragraph it will, in most circumstances, ensure that the deck seal is located in the vicinity of the cargo pump room and indeed no objection would be taken in principle to placing the deck seal either on top of the pumproom or even inside it, although the latter location has its drawbacks since the pumproom has to be ventilated for some minutes before entry as indicated in paragraph 5, 15, 1.7.2.

Paragraph 5, 15, 7.5.7. The water seal referred to in 7.5.5 is to be capable of being supplied by two separate pumps, each of which is to be capable of maintaining an adequate supply at all times. (Regulation 62.10.3).

Very often the water seal is supplied by the same pumps as supply the scrubber (*See* Fig. 4). However, no objection is taken to the deck seal being supplied independently provided the source of supply is from two pumps. The deck seal can, for example, be supplied from the hydrophore sanitary system.

Paragraph 5, 15, 7.5.8. The arrangement of the seal and its associated fittings is to be such that it will prevent backflow of hydrocarbon vapours and will ensure the proper functioning of the seal under operating conditions. (Regulation 62.10.4).

Turning to Fig. 4 for a moment it will be seen that the deck seal has two comments appended. The first concerns the height of the inert gas inlet pipe which should be capable of being filled with water from the reservoir to a height sufficient to prevent a backflow of gas from the cargo tanks, that is equivalent to 0.24 kgf/cm^2. The second comment refers to the water supply pipe which should have a non-return valve or loop fitted adjacent to the deck seal. The open end of the supply pipe should also be submerged to below the level at which the low level alarm operates. See 5, 15, 7.7.7 (g). The reason for these safeguards is to prevent the possible passage of hydrocarbon gases back into a safe space.

It may be thought that with all the safeguards incorporated into an inert gas system the return of hydrocarbon gases would be unlikely, but an inert gas system contravenes the intention of the Rules in that there should be no direct connection, unless foolproof, between the cargo oil arrangements and the ordinary pumping arrangements or, taking a simple view, between the safe and dangerous areas, and yet here is a system which, although fitted as a safeguard, is connected directly from the cargo tanks into the machinery space.

To illustrate this point an explosion occurred in the blower room of a tanker due to the fact that the inert gas system on this particular ship had not been used for three weeks and all the alarms had either been shut off or bypassed. As a result the deck seal had no water in it and hydrocarbon gases had come back from the cargo tanks, into the deck seal, thence into the water supply pipe and back to the scrubber and eventually leaked out through broken blower shaft seals into the blower room where the explosion took place—and, it should be emphasised, the blower room is supposed to be a safe space. The blower starters were probably the source of ignition.

Paragraph 5, 15, 7.5.9. Provision is to be made to ensure that the water seal is protected against freezing, in such a way that the integrity of seal is not impaired by over heating. (Regulation 62.10.5).

Fitting a steam coil is the usual way of complying with this paragraph. If the internal surfaces of the deck seal are lined with a protective coating the heating coil should be kept well clear.

Paragraph 5, 15, 7.5.10. A water loop or other approved arrangement is also to be fitted to each associated water supply and drain pipe and each venting or pressure-sensing pipe leading to gas safe spaces. Means are to be provided to prevent such loops from being emptied by vacuum. (Regulation 62.10.6).

This paragraph refers to the water supply to the deck seal only and the drain overboard from it. The means of preventing such loops from being emptied by vacuum refers to the possibility of siphoning taking place and this is prevented by fitting an air pipe, or equivalent, to the top of the loop and leading the outlet to the open. Examples of these loops can be seen in the overboard discharges from the deck seal and scrubber in Fig. 4. The pressure devices are usually fitted in conjunction with transducers which would prevent the passage of gases.

Paragraph 5, 15, 7.5.11. The deck water seal and all loop arrangements are to be capable of preventing return of hydrocarbon vapours at a pressure equal to the test pressure of the cargo tanks. (Regulation 62.10.7—Need not be complied with for existing I.G. systems. *See* 62.20.1).

This indicates that the water loops mentioned in the previous paragraph should be at least 2.4 metres high depending whether they are located on the suction side or discharge side of the blowers.

The capability of the deck seal itself to prevent the return of hydrocarbon gas is achieved by the fact that if the hydrocarbon gases should try to return they will press down upon the water in the deck seal and force the water up the inert gas inlet pipe to a height of 2.4 metres. See Fig. 4 for note in this respect. Before this column of water reaches 2.4 metres two things should have happened. Firstly the pressure vacuum valve adjacent to the vent mast should have lifted and relieved the pressure. This is often set to about 1400 mm water gauge. However, this valve could be stuck due to the hydrocarbon vapours having condensed out on to the valve and seat forming a gluey deposit. If it doesn't lift then the only other device to prevent the return of gases and also to protect the cargo tanks from overpressure, is the oil filled pressure vacuum breaker referred to in 5, 15, 7.6.3.

Paragraph 5, 15, 7.5.12. The second non-return device is to be a non-return valve or equivalent capable of preventing the return of vapours or liquids and fitted forward of the deck water seal required in 7.5.5. It is to be provided with positive means of closure. As an alternative to positive means of closure, an additional valve having such means of closure may be provided forward of the non-return valve to isolate the deck water seal from the inert gas main to the cargo tanks. (Regulation 62.10.8).

This non-return valve is usually a flap valve with a weight loaded external handle to give a nominal closing pressure and to keep the valve on its seat during the ship's motion in heavy weather, but not sufficient to prevent the inert gas from passing through.

Paragraph 5, 15, 7.5.13. As an additional safeguard against the possible leakage of hydrocarbon liquids or vapours back from the deck main, means are to be provided to permit this section of the line between the valve having positive means of closure referred to in 7.5.12 and the valve referred to in 7.5.3 to be vented in a safe manner when the first of these valves is closed. (Regulation 62.10.9—Need not be complied with for existing I.G. systems. *See* 62.20.1).

This vent valve is shown in Fig. 4 just foward of the bulkhead valve and should be opened when the inert gas system is shut down to prevent any leakage past the non-return valve and deck seal from building up any pressure in the inert gas line between the regulating valve referred to in 5, 15, 7.5.3 and these non-return devices. It is debatable whether it would not be better to fit the vent between the non-return valve and the deck main isolating valve or even between the non-return valve and the deck seal since there would then be no question of building up a pressure between the deck seal and the regulating valve.

Paragraph 5, 15, 7.5.14. The inert gas main may be divided into two or more branches forward of the non-return devices required by 7.5.5. (Regulation 62.11.1).

The word "branches" in this paragraph is possibly misleading. What is meant is that for some product/chemical carriers it is sometimes necessary for the purposes of cargo segregation, between either incompatible cargoes or cargoes which require high levels of purity, to have more than one deck main after the non-return valve. Each main thereafter should have its own isolating valve.

Paragraph 5, 15, 7.5.15. The inert gas supply mains are to be fitted with branch piping leading to each tank. Branch piping for inert gas is to be fitted with either stop valves or equivalent means of control for isolating each tank. Where stop valves are fitted, they are to be provided with locking arrangements, which are to be under the control of a responsible ship's officer. (Regulation 62.11.1).

The "equivalent means of control for isolating each tank" referred to above normally involves blanks and Fig. 12 shows a spade blank, made of Tufnol, which, due to its shape, cannot be fitted unless the hatch lid is open and the hatch lid cannot therefore be closed unless the blank is removed.
Assuming the tank has been gas freed before the blank is fitted, this means that it will be reasonably safe for men to work in that tank not only with an assured air supply but also with the safeguard that no inert gas can enter. Furthermore, with the hatch having to remain open because of the blank, there is little possibility of the blank being left in by mistake afterwards and, in any case, paragraph 5, 15, 4.2.4 of the Rules which states that "means are to be provided to prevent any tank being subjected to excessive pressure or vacuum during any phase of the cargo handling or ballasting operations", will have been complied with.

However, it should be emphasised that in many instances, as shown in Fig. 4 and paragraphs 5, 15, 4.2.1 and 5, 15, 7.6.1 of the Rules, the pipe lines on deck delivering inert gas to the cargo tanks can also be used for venting the tanks. In these circumstances, if the inert gas isolating valve at the tank is closed, or an external blank is fitted, the tank vent line or breathing passage is shut off and paragraph 5, 15, 4.2.4 would not be complied with. Fortunately, paragraph 5, 15, 7.5.17 now indicates that cargo tanks need only be protected against overpressure or vacuum caused by thermal variation allowing, for example, the tanks to breathe out during a hot day and to breathe in during a cool night. This is usually effected by small pressure-vacuum valves. More radical changes of pressure or vacuum due to loading, unloading or ballasting operations are now operational but some protection is given by the high and low pressure alarms referred to in paragraphs 5, 15, 7.7.7. (h) and (i). However, the breathing valves do not fully take into

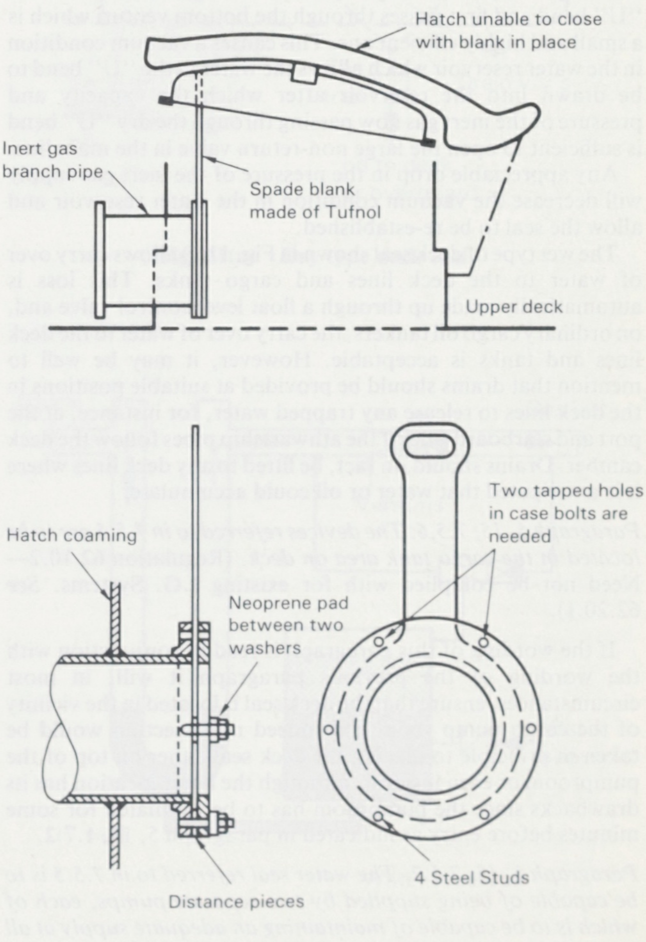


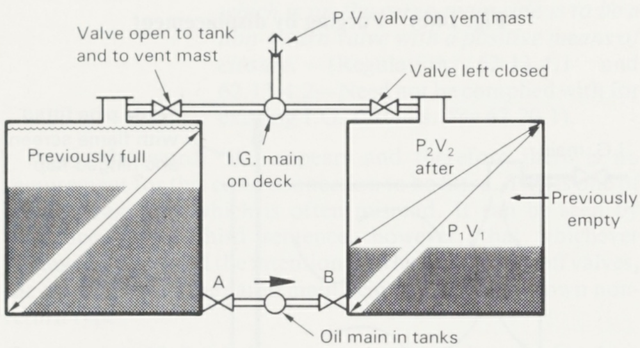
Fig. 12 Spade blank flange arrangement

consideration inadvertent conditions arising such as illustrated in a simplified manner in Fig. 13 where a full tank is put into communication with an empty tank which could create a vacuum or overpressure condition, possibly with disastrous results. This has occured several times and its avoidance is very much a matter for care and attention by the crew.

Where blanks are fitted in the inert gas branch lines external to the hatch coaming, one means of ensuring that a blank has not been left in place is to have a box with numbered slots so that an easy check can be made.

The locking arrangements referred to usually mean chains and padlocks the intention being, on most occasions, for the valves to be locked open to keep all tanks common with the inert gas main or vent main and, on other occasions, to lock the valves shut when, for instance, someone had gone into a cargo tank to effect repairs or to make an inspection, perhaps associated with the crude oil washing process.

To illustrate in a simplified manner how an overpressure or vaccum condition can be created just by dropping the contents of a full tank either "into" or "from" a Non-Vented tank (see Rules 5.15, 7.15)



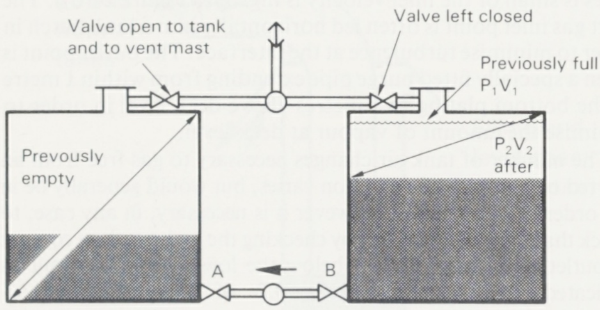
Overpressure condition if valves "A" and "B" inadvertently opened

Before: $P_1 = 1ATM \quad V_1 = 10000$

After: $P = ? \quad V_2 = 6000\ m^3$

$P_1V_1 = P_2V_2 \therefore P = \frac{1 \times 10000}{6000} = 1.67ATM$

$\therefore$ The pressure has nearly doubled and the tank may rupture



Vacuum condition if valves "A" and "B" inadvertently opened

Before: $P_1 = 1ATM \quad V_1 200\ m^3$

After: $P_2 = ? \quad V_2 = 4000\ m^3$

$P_1V_1 = P_2V_2 \therefore P_2 = \frac{1 \times 200}{4000} = 0.05ATM$

$\therefore$ Almost a full vacuum condition and the tank may collapse

Fig. 13 Overpressure/underpressure conditions

Paragraph 5, 15, 7.5.16. In combination carriers, the arrangement to isolate the slop tanks containing oil or oil residues from other tanks is to consist of blank flanges which will remain in position at all times when cargoes other than oil are being carried except as provided for in 5, 15, Sub-Section 1.9. (Regulation 62.11.2.2).

So far as blanking off the inert gas from the cargo holds when carrying dry cargoes is concerned the blank flanges referred to should generally be fitted in the inert gas main forward of the branch lines to the slop tanks and aft of the first branch to a cargo tank. However, the inert gas arrangements on combination carriers vary and, depending upon whether some cargo tanks require to be inerted during dry cargo voyages, it may be necessary to also fit blanks in other positions. Each case has to be carefully considered. More detailed operational aspects for combination carriers are given in Section 7 of the Inert Gas Guidelines mentioned in the Introduction.

Paragraph 5, 15, 7.5.17. Means are to be provided to protect cargo tanks against the effect of overpressure or vacuum caused by thermal variations when the cargo tanks are isolated from the inert gas mains. (Regulation 62.11.3—Need not be complied with for existing I.G. systems. *See* 62.20.1).

See paragraph 5, 15, 7.5.15 for comments.

Paragraph 5, 15, 7.5.18. Piping systems are to be so designed as to prevent the accumulation of cargo or water in the pipelines under normal conditions. (Regulation 62.11.4—Need not be complied with for existing I.G. systems. *See* 62.20.1).

This is generally referring to deck pipes where they follow the camber of the decks and create pockets of liquid at port and starboard sides.

In the case of combination carriers with their large deck hatches, sloshing within the hatch coamings can cause the inert gas lines to be filled with cargo oil and, in order to combat this, the inert gas line is often looped upwards to a height just below the line of the hatch cover before it enters the tank.

Paragraph 5, 15, 7.5.19. Suitable arrangements are to be provided to enable the inert gas main to be connected to an external supply of inert gas. (Regulation 62.11.5).

In the event that a tanker's inert gas system has broken down these arrangements should enable a supply of inert gas to be connected either from another tanker or from shore.

In order to assist shipowners, shipbuilders and Administrations in determining acceptable design requirements for compliance with the above the International Chamber of Shipping (ICS) and the Oil Companies International Marine Forum (OCIMF), in consultation with the International Association of Classification Societies (IACS) have prepared the following guidance:

"The arrangements should consist of a 250 mm bolted flange conforming to ANSI 150*, isolated from the inert gas main by a valve and located forward of the non-return valve referred to in 5, 15, 7.5.12. A blank flange should be fitted at the open end when not in use".

It should be noted that the above definition is for guidance only and compliance could not be insisted on.

Sub-Section VENTING ARRANGEMENTS

Paragraph 5, 15, 7.6.1. The arrangements for the venting of all vapours displaced from the cargo tanks during loading and ballasting are to comply with Section 4 and are to consist of either one or more mast risers, or a number of high velocity vents. The inert gas supply mains may be used for such venting. (Regulation 62.12—Need not be complied with for existing I.G. systems. *See* 62.20.1 and 62.20.2).

*ANSI 150: American National Standards Institute Specification B-16:5 Class 150 (British Standards equivalent 1560 Class 150).

This paragraph is generally self explanatory and perhaps the only comment to make is that the high velocity vents should be of an approved type as per 5, 15, 4.5.4. More is written about these high velocity vents in Part II of this paper.

Paragraph 5, 15, 7.6.2. The arrangements for inerting, purging or gas freeing of empty tanks as required in 7.2.3 are to be such that the accumulation of hydrocarbon vapours in pockets formed by the internal structural members in a tank is minimized and that:

Sub-Paragraph (a) *on individual cargo tanks the gas outlet pipe, if fitted, is to be positioned as far as practicable for the inert gas/air inlet and in accordance with 5, 15, Section 4. The inlet of such outlet pipes may be located either at deck level or at not more than 1 mm above the bottom of the tank;* (Regulation 62.13.1—Need not be complied with for existing I.G. systems. *See* 62.20.1 and 62.20.2).

Note the words "if fitted" in the first line which indicates that purge pipes do not have to be fitted although it would be expected that alternative means of purging the cargo tanks with either inert gas or air would be provided, the outlets being either through the cargo lines at the manifold loading stations or through the venting system, such as the mast risers or high velocity vents. [*See also* paragraph 5, 15, 7.2.3(d)].

Figures 14(a), (b), (c), and (d) show arrangements of purge pipes referred to in this paragraph.

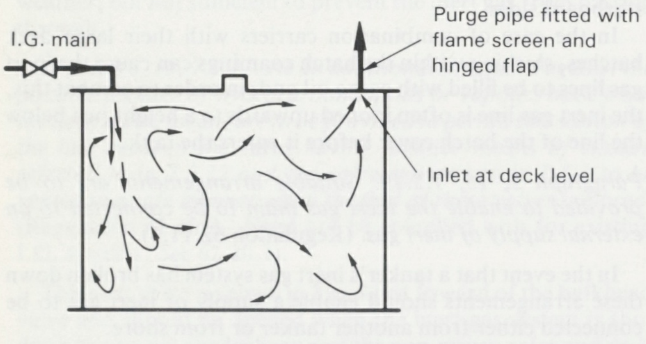


Fig. 14(a) Purging by dilution

Gases are both introduced and vented from the top of the tank. This is the simplest arrangement. Gas replacement is by the dilution method. The incoming gas should always enter the tank in such a way as to achieve maximum penetration and thorough mixing throughout the tank. Gases can be vented through a purge or vent pipe on each tank or through a common vent main.

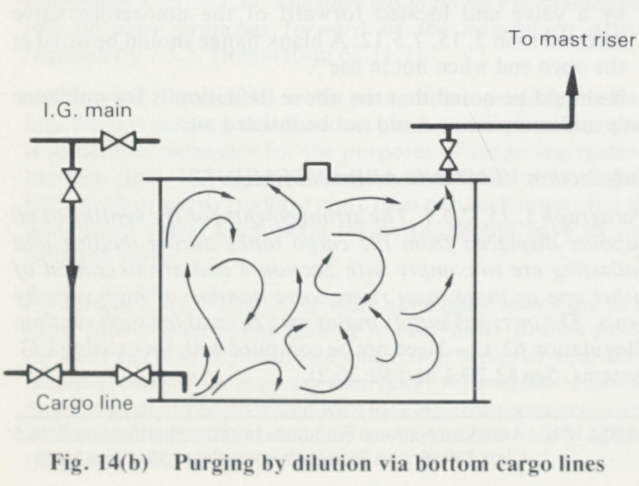


Fig. 14(b) Purging by dilution via bottom cargo lines

Gas is in introduced at the bottom of the tank and vented from the top. Gas replacement is by the dilution method. This arrangement introduces the gas through a connection between the inert gas deck main (just forward of the mechanical non-return valve) and the bottom cargo lines (*see* Fig. 4). A special fixed gas-freeing fan may also be fitted. Exhaust gas may be vented through individual purge or vent pipes or, if valves are fitted to isolate each cargo tank from the inert gas main, through this main to the mast riser or high velocity vent.

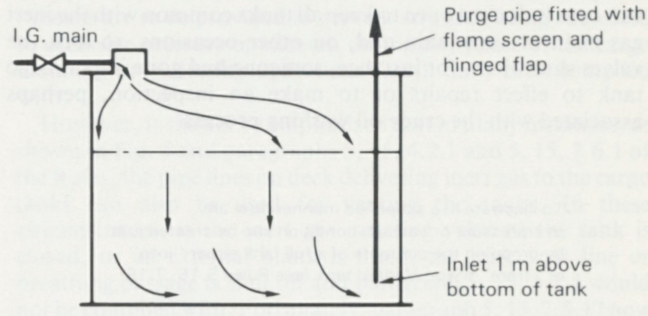


Fig. 14(c) Purging by displacement

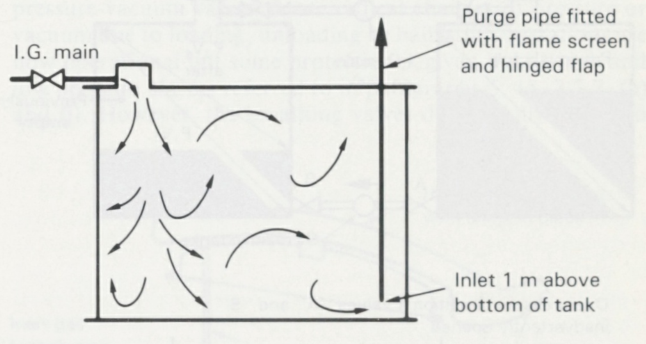


Fig. 14(d) Purging by dilution

Gas is introduced at the top of the tank and discharged from the bottom. This arrangement permits the displacement method (*see* Figure 14(c)) although the dilution method may predominate if the density difference between the incoming and existing gases is small or the inlet velocity is high (*see* Figure 14(d)). The inert gas inlet point is often led horizontally into a tank hatch in order to minimise turbulence at the interface. The outlet point is often a specially fitted purge pipe extending from within 1 metre of the bottom plating to 2 metres above deck level in order to minimise the amount of vapour at deck level.

The number of tank air changes necessary to gas free from an inerted or non-inerted condition varies, but would generally be in the order of 2 to 4 times. However it is necessary, in any case, to check that the tank is gas free by checking the hydrocarbon content at outlet to ensure that it is below the lower explosive limit as indicated in Regulation II-2 59.2 of the 1981 SOLAS Amendments.

Sub-Paragraph (b) *the cross sectional area of such gas outlet pipes referred to in (a) is to be such that an exit velocity of at least 20 m/s can be maintained when any three tanks are being simultaneously supplied with inert gas. Their outlets are to extend not less than 2 m above deck level;* (Regulation 62.13.2—Need not be complied with for existing I.G. systems. *See* 62.20.1 and 62.20.2).

Note that the height of purge pipe outlets is the same as the height of outlets from "breathing" pressure vacuum valves and high velocity vent heads as per 5, 15, 4.5.3 of the Rules.

Sub-Paragraph (c) *Each gas outlet referred to in (b) is to be fitted with suitable blanking arrangements.* (Regulation 62.13.3).

The blanking arrangements are usually in the form of a hinged lid and, as indicated in the comments on paragraph 5, 15, 7.2.3(d), the purge pipes of inerted and non-inerted tankers would also require to be fitted with a device to prevent the passage of flame.

Sub-Paragraph (d) *if a connection is fitted between the inert gas supply mains and the cargo piping system, arrangements are to be made to ensure an effective isolation having regard to the large pressure difference which may exist between the systems. This is to consist of two shut-off valves with an arrangement to vent the space between the valves in a safe manner or an arrangement consisting of spool-piece with associated blanks. The valve separating the inert gas supply main from the cargo main and which is on the cargo main side is to be a non-return valve with a positive means of closure.* (Regulation 62.13.4.1 and 62.13.4.2—Need not be complied with for existing I.G. systems. *See* 62.20.1).

Again the word "if" appears and therefore there is no requirement for this cross connection to be fitted. This is one of those paragraphs which is often misread. It can be seen by reference to the third sentence, however, that whichever arrangement is fitted the intention is to have two shut-off valves, the one nearest to the cargo main being of the screw down non-return type.

Paragraph 5, 15, 7.6.3. One or more pressure-vacuum breaking devices are to be provided to prevent the cargo tanks from being subject to: (Regulation 62.14.1—Note that the wording of this paragraph has been slightly amended by the 1983 Amendments to SOLAS 1974).

Figure 15 shows a liquid filled pressure/vacuum breaker. This is fitted in addition to the P.V. valve(s) in the system, the reason being that the P.V. valves, or high speed vents, can become coated with oil either from the vapour or oil overflow and become sluggish in operation or even stick closed. In such circumstances the P. V. breaker becomes the final safeguard against the cargo tanks either rupturing or collapsing by either expelling the liquid onto the deck in an overpressure condition or by sucking the liquid into the inert gas line, or tanks, in an underpressure or vacuum condition. Should such a circumstance occur, the breaker must be refilled before the I.G. plant is put back into operation. The liquid should, of course, be of the non-freezing type such as a mixture of glycol and water and it is important that filling the P.V. breaker is done when the cargo tanks are at atmospheric pressure, for example when the hatch lids are open, otherwise wrong operational settings will result.

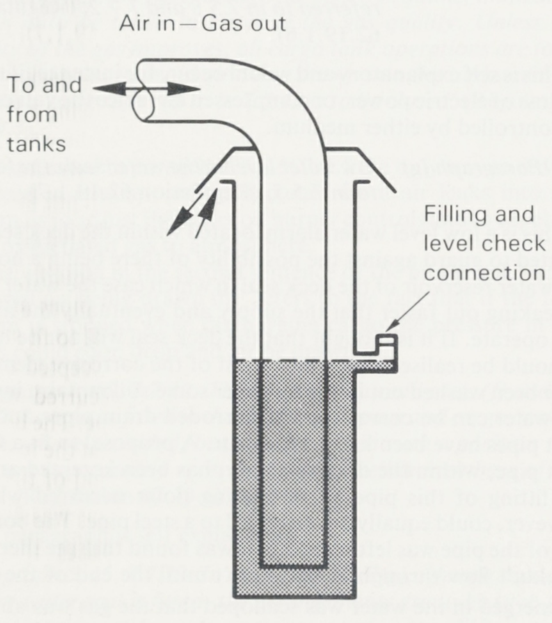


Fig. 15 Pressure vacuum breaker 5, 15, 7.6.3

Sub-Paragraph (a) *a positive pressure in excess of the test pressure if the cargo were to be loaded at the maximum rated capacity and all other outlets are left shut. . .*(Regulation 62.14.1.1.—Note that the wording of this paragraph has been slightly amended by the 1983 SOLAS Amendments to SOLAS 1974).

The positive pressure referred to should not normally exceed 0.24 kgf/cm^2. It would appear from the wording of this paragraph that the design rate gas discharge capacity of the P. V. breaker could govern the loading rate of the tanker, assuming other P. V. valves to be stuck closed. Since loading rates are variable, a high gas pressure alarm is provided in the inert gas main as indicated in 5, 15, 7.7.7 (j) and should this operate the loading rate would have to be reduced.

Sub-Paragraph (b) *a negative pressure in excess of 700 mm water gauge if cargo were to be discharged at the maximum rated capacity of the cargo pumps and the inert gas blowers were to fail. Such devices shall be installed on the I.G. main unless they are installed in the venting system required by Reg. 59.1.1 or on individual tanks.* (Regulation 62.14.1.2—Note that the last sentence has been added by the 1983 Amendments to SOLAS 1974).

If the blowers were to fail with the tank full the vacuum effect would be immediate whereas with a tank half empty, or empty, the vacuum effect would take some time to be felt due to the expansive capability of the tank atmosphere.

As a general comment and with regard to the references to the 1983 SOLAS Amendments it is hoped that these amendments will eventually be reflected in the Society's Rules. There are also other subtle amendments in the 1983 SOLAS Amendments which have not been detailed in order to avoid confusion but have been taken account of in this paper.

Paragraph 5,15, 7.6.4. The location and design of the devices referred to in 7.6.3 are to be in accordance with 5, 15, Section 4. (Regulation 62.14.2—Need not be complied with for existing I.G. systems. *See* 62.20.1 and 62.20.2).

P.V. breakers should be located not less than 10 m from air intakes and openings to accomodation, sources of ignition etc, as per 5, 15, 4.5.1.

In the case of combination carriers (OBO and ORE/OIL) the P.V. breaker, whilst complying with the above, should preferably be located aft of the blanking arrangements required for the dry cargo condition, see 5, 15, 7.5.16, so that the slop tanks are still protected for overpressure and underpressure conditions.

By virtue of a P.V. breaker's vertical tubular design the height of the opening would normally be about 2 m above deck but, in any case, this would not be important compared to the operation of the breaker in its emergency operating mode.

Sub-Section 7.7 INSTRUMENTATION AND ALARMS

Paragraph 5, 15, 7.7.1. Means are to be provided for continuously indicating the temperature and pressure of the inert gas at the discharge side of gas blowers, whenever the gas blowers are operating. (Regulation 62.15).

This implies a thermometer in a pocket and a pressure gauge, or equivalent. *See also* section 3.14 of the Inert Gas Guidelines.

Paragraph 5,15, 7.7.2. Instrumentation is to be fitted for continuously indicating and permanently recording, when the inert gas is being supplied:

(a) the pressure of the inert gas supply mains forward of the non-return devices required by 7.5.5, and

(b) the oxygen content of the inert gas in the inert gas supply mains on the discharge side of the gas blowers. (Regulation 62.16.1).

This paragraph is considered self explanatory.

Paragraph 5, 15, 7.7.3. The devices referred to in 7.7.2 are to be placed in the cargo control room where provided. But where no cargo control room is provided, they are to be placed in a position easily accessible to the officer in charge of cargo operations. (Regulation 62.16.2).

This paragraph is considered self explanatory.

Paragraph 5, 15, 7.7.4. In addition to 7.7.2 meters are to be fitted: (Regulation 62.16.3).

Sub-Paragraph (a) *in the navigating bridge to indicate at all times the pressure referred to in 7.7.2(a) and the pressure in the slop tanks of combination carriers, whenever those tanks are isolated from the inert gas supply main, and*

It is sometimes assumed that the inert gas pressure measurement required by 5, 15, 7.7.2(a) is sufficient to indicate the pressure in the slop tanks. However, if the slop tanks are isolated by valves or blanking arrangements external to the hatch coamings then the pressure in the slop tanks cannot be transmitted via the inert gas main and it would therefore be necessary to fit pressure indicator lines direct from the slop tanks to the navigating bridge.

Sub-Paragraph (b) *in the machinery control room or in the machinery space to indicate the oxygen content referred to in 7.7.2(b).*

This sub-paragraph is considered self explanatory.

Paragraph 5, 15, 7.7.5. Portable instruments for measuring oxygen and flammable vapour concentration are to be provided. In addition, suitable arrangement is to be made on each cargo tank that the condition of the tank atmosphere can be determined using these portable instruments. (Regulation 62.17).

Since paragraph 5, 15, 5.3.1 of the Rules states that closed sounding arrangements should be provided when tankers are fitted with inert gas systems this appears to be in contradiction to the use of portable instruments which often require inerted tanks to be opened for dipping, measuring and sampling. These, however, are operations which are necessary in connection with cargo handling operations and safe crude oil washing procedures and are acceptable provided a positive inert gas pressure, which may be almost atmospheric, is maintained within the cargo tanks.

Paragraph 5, 15, 7.7.6. Suitable means are to be provided for the zero and span calibration of both fixed and portable gas concentration measurement instruments, referred to in 7.7.2, 7.7.4 and 7.7.5. (Regulation 62.18).

The word "span" means that small bottled samples of each type of gas which the instruments may require to measure should be carried on board so that an accurate calibration can be made. Oxygen "span" gas does not require to be carried since the oxygen meter can be calibrated to show 21% oxygen by passing air through the meter.

Paragraph 5, 15, 7.7.7. Audible and visual alarms are to be provided to indicate: (Regulation 62.19.1).

Sub-Paragraph (a) *low water pressure or low water flow rate to the flue gas scrubber as referred to in 7.3.1.*

If the water flow is reduced, or stops, the temperature in the scrubber will rise and, as seen from 5, 15, 7.7.9, the blowers will stop and the bulkhead valve will close. (*See also* paragraph 5, 15, 7.5.3 for comment).

Sub-Paragraph (b) *high water level in the flue gas scrubber as referred to in 7.3.1,* (Regulation 62.19.12).

See paragraph 5, 15, 7.5.3 for comment.

Sub-Paragraph (c) *high gas temperature as referred to in 7.7.1,* (Regulation 62.19.1.3).

The alarm is usually set to 65°C (149°F), bearing in mind that the temperature should not present a source of ignition or melt off any pipeline or scrubber lining. (*See also* 5, 15, 7.5.3). As can be seen from 5, 15, 7.7.9 the blowers will stop and the bulkhead valve will close, usually at about 75°C (167°F).

Sub-Paragraph (d) *failure of the inert gas blowers referred to in 5,15,7.7.4,* (Regulation 62.19.1.4).

As indicated in 5, 15, 7.7.10 the bulkhead valve will close.

Sub-paragraph (e) *oxygen content in excess of 8 per cent by volume as referred to in 7.7.2(b).*(Regulation 62.19.1.5).

If the oxygen content should rise above 8% then the automatic combustion control, mentioned at the end of 5, 15, 7.2.1, is wrong and has to be corrected as soon as possible. However, the paragraph under consideration does not say that the plant should be shut down since inert gas having an oxygen content of 8%, 10% or 12% is still better than supplying air with 21% oxygen.

Sub-Paragraph (f) *Failure of the power supply to the automatic control system for the gas regulating valve and to the indicating devices as referred to in 7.5.3 and 7.7.2,* (Regulation 62.19.1.6).

This is self explanatory and would occur, for instance, if there is a loss of electric power, or compressed air, since the valves can be controlled by either medium.

Sub-Paragraph (g) *low water level in the water seal as referred to in 7.5.5,* (Regulation 62.19.1.7).

This is a low level water alarm located within the deck seal. It is fitted to guard against the possibility of there being a hole in the water reservoir of the deck seal in which case the water may be leaking out faster that the supply and eventually the alarm will operate. If it is thought that the deck seal will not be holed it should be realised that whilst most of the corrosive elements have been washed out in the scrubber some still remain, indeed sea water can be corrosive, and corroded drain pipes and gas inlet pipes have been known to occur. A proposal to fit a GRP inlet pipe, within the deck seal only, has been accepted and in the fitting of this pipe an interesting point occurred which, however, could equally have applied to a steel pipe. The bottom end of the pipe was left plain and it was found that the inert gas wouldn't flow through and it wasn't until the end of the pipe submerged in the water was scalloped that the gas was able to pass through. The incident perhaps emphasises the low pressures at which inert gas systems operate.

Sub-Paragraph (h) *gas pressure less than 100 mm water gauge as referred to in 7.7.2(a). The alarm arrangement is to be such as to ensure that the pressure in slop tanks in combination carriers can be monitored at all times, and,* (Regulation 62.19.1.8).

This alarm would appear to be set very low but as indicated in Fig. 7 the inert gas itself operates at a low pressure. *See also* 5, 15, 7.7.4 regarding slop tank pressure measurement.

Sub-Paragraph (j) *high gas pressure as referred to in 7.7.2(a).* (Regulation 62.19.1.9).

If this alarm operates it shows that either the gas regulating valve has been set incorrectly or, more importantly, that an accumulation of pressure is occurring in the cargo tanks perhaps because the loading rate is excessive or the flame screens or arresters are getting blocked and not allowing the gases to escape. In these circumstances, unless the loading rate is reduced, the only safeguard to prevent overpressuring the cargo tanks is for the pressure/vacuum breaker, referred to in 5, 15, 7.6.3, to operate.

Paragraph 5, 15, 7.7.8. In the system with gas generators audible and visual alarms are to be provided in accordance with (a), (c), (e) to (j) of 7.7.7 and additional alarms to indicate:

(a) insufficient fuel oil supply,

(b) failure of the power supply to the generator,

(c) failure of the power supply to the automatic control system for the generator. (Regulation 62.19.2).

This paragraph is considered self explanatory.

Paragraph 5, 15, 7.7.9. Automatic shut-down of the inert gas blowers and gas regulating valve is to be arranged on predetermined limits being reached in respect of (a), (b) and (c) of 7.7.7. (Regulation 62.19.3).

These conditions have been dealt with in the relevant sub-paragraphs.

Paragraph 5, 15, 7.7.10. Automatic shut-down of the gas regulating valve is to be arranged in respect of 7.7.7(d). (Regulation 62.19.4)

This condition has been dealt with in the relevant sub-paragraph.

Paragraph 5, 15, 7.7.11 In respect of 7.7.7(e), when the oxygen content of the inert gas exceeds 8 per cent by volume, immediate action is to be taken to improve the gas quality. Unless the quality of the gas improves, all cargo tank operations are to be suspended so as to avoid air being drawn into the tanks and the isolation valve referred to in 7.5.12 is to be closed. (Regulation 62.19.5).

As indicated in paragraph 5, 15, 7.2.5 the only way to reduce the oxygen content, unless it comes from air leaks into the system, is to adjust the boiler oil burner controls and to expel the over oxygenated gas to atmosphere.

The wording in the second sentence of the paragraph "so as to avoid air being drawn in" refers to the fact that if the I.G. system is stopped when discharging cargo the pressure in the tanks could soon drop and air be drawn in.

Paragraph 5, 15, 7.7.12. The alarms required in (e), (f) and (h) of 7.7.7 are to be fitted in the machinery space and cargo control room, where provided, but in each case in such a position that they are immediately received by responsible members of the crew. (Regulation 62.19.6).

This paragraph is self explanatory.

Paragraph 5, 15, 7.7.13. In respect of 7.7.7(g), where a semi-dry or dry water seal is fitted, the arrangements are to be such that the maintenance of an adequate reserve of water will be ensured at all times and that the water seal will be automatically formed when the gas flow ceases. The audible and visual alarm on the low level of water in the water seal is to operate when the inert gas is not being supplied. (Regulation 62.19.7).

Whenever the tanker is carrying oil or is not in a gas free condition and, even though the inert gas system may have been shut down and isolated from the cargo tanks, the water supply to the deck seal should be maintained as a safeguard against hydrocarbon gases coming back from the cargo tanks.

It should be noted that the dry type of deck seal referred to in paragraph 5, 15, 7.5.5 and shown in Fig. 11(a), runs dry when inert gas is being supplied and therefore the low water level alarm would require to be by-passed during this time and then re-activated when the seal has eventually been re-established.

Paragraph 5, 15, 7.7.14. An audible alarm system independent of that required in 7.7.7(h) or automatic shut down of cargo pumps is to be provided to operate on predetermined limits of low pressure in the inert gas mains being reached. (Regulation 62.19.8—Need not be complied with for existing I.G. systems. *See* 62.20.1).

In addition to the alarm referred to in 5, 15, 7.7.7(h) this second low pressure alarm or automatic shut down of the cargo pumps should take place at 50 mm water gauge. The period between the two alarms will vary according to how full the cargo tanks are. The option of automatic shut down of the cargo pumps can result in boiler safety valves blowing since the steam load will suddenly be removed and it could also affect the automatic combustion control system. This option is usually found only on existing inert gas installations.

It is important that the 100 mm and 50 mm pressure sensors should be led via two separate tubes off the inert gas main, after the deck isolating valve referred to in 5, 15, 7.5.12, since the tubes are small and can easily become blocked.

Paragraph 5, 15, 7.7.15. Detailed instruction manuals are to be provided on board, covering the operations, safety and maintenance requirements and occupational health hazards relevant to the inert gas system and its application to the cargo tank system. The manuals are to include guidance on procedures to be followed in the event of a fault or failure of the inert gas system. (Regulation 62.21).

Page 44 of the Inert Gas Guidelines referred to in the Introduction gives a good description of what a Manual should contain. It should be noted that "Instruction Manuals" for inert gas systems are not required to be "Approved" as, for example, the Crude Oil Washing Manual requires to be and the Society has so far declined to approve any without an internationally agreed specification, such as that for the COW manual.

Sub-Section 7.7 *INSTALLATION AND TESTS*

Paragraph 5, 15, 7.8.1. The inert gas system, including alarms and safety devices, is to be installed on board and tested under working conditions to the satisfaction of the Surveyors. (Regulation 62—No equivalent).

Appendix "B" shows a suggested test programme which has been included for general guidance. *See also* Part 1, Chapter 3 Sub-Section 2.2.21 of the Society's Rules regarding Annual Surveys and Part 1, Chapter 3, Section 17 for Special Surveys.

This completes comments on the Society's Rules for inert gas systems and most of the points on a typical flue gas systems have now been made.

However, futher mention should be made of the overboard drains from the scrubber and deck seal.

1.1.10 Scrubber and Deck Seal Drains

These are dealt with by the International Conventions Department, Load Line Section, since they come under the definition of scuppers as per Part 3, Chapter 12, Section 4 of the Rules rather than being considered part of a totally enclosed piping system.

Regarding the drain from the scrubber, better known as the effluent overboard discharge line, it will be realised from what has been said earlier that this line is subject to high corrosive attack from sulphuric acid and erosive attack from the solid particles washed out from the flue gases and it is therefore not surprising to find that in many cases this pipe is very quickly wasted away within months of installation. Various choices of materials and arrangements have been used but undoubtedly the best pipe to use is one made of GRP. This, however, has the major drawback of having a low melting temperature and is therefore susceptible to fire damage and subsequent flooding could occur since it is likely that the metallic shipside valve will also have been corroded and cannot be tightly shut off.

In the early days GRP effluent piping was not accepted except when fitted inside a metallic pipe, which, in practice, is very difficult to fit and therefore very uncommon. However, in order to take advantage of the benefits of GRP in this situation and specifically with regard to the scrubber effluent overboard discharge pipe, a GRP pipe is now acceptable within the machinery space provided a "fail to close" compressed air or hydraulic actuated butterfly valve, or equivalent, is fitted at the shell with a back-up non-return flap valve of stainless steel having intervening spool pieces with testing drains to establish if the valves are tight and to facilitate removal of the non-return valve, if required. A sketch of the arrangement is shown in Fig. 16.

Apart from the above, some of the various choices of materials and arrangements for inert gas scrubber discharge pipes which are considered acceptable, until such time as experience and feed-back indicate the superiority or otherwise of other arrangements and materials, are as follows:

(i) distance piece from shell to valve to have thickness of D/10 with a maximum of 20 mm and a minimum of 10 mm. the distance piece to be lined with rubber, GRP or stainless steel.

(ii) It is recommended that the shipside valve be of the diaphragm type, lined with rubber or other suitable material, geared to the nearest working platform. However, a N.R. valve, or butterfly valve, suitably protected, will also be considered.

(iii) The overboard discharge pipe inboard of the valve extending to the scrubber (or upper deck, whichever is the lower) to be:

(a) Steel pipe 9 mm thick, rubber lined.

(b) 316 stainless steel pipe 6 mm thick.

(c) GRP pipe of substantial thickness not less than 12.5 mm, encased between scrubber tank (or upper deck) and the shell valve, in a steel pipe 6.5 mm thick, so arranged as to provide an adequate air insulation space. Distance rings to be fitted about 3 metres apart.

In the case of iii(c), a ship's side valve need not be fitted provided the GRP discharge pipe is encased in a steel pipe of thickness 12.5 mm from shell to a height .02L above the L.W. and 9 mm above .02L. Special attention would require to be given to the jointings and sealing of the GRP pipe to obviate any leakage of effluent into the air space.

Regarding the diaphragm valve referred to in (ii) above, the overboard discharge pipes from the scrubber or deck seal are the only locations where diaphragm valves have been accepted as shipside valves. This is because they can be rubber lined throughout and therefore offer good resistance to the corrosive and erosive effluents, especially from the scrubber.

As can be seen from Fig. 17 the reason diaphragm valves are not normally accepted as shipside valves is that should the diaphragm split there is no way of shutting off any influx of water.

All shipside valves should be made of ductile material. See part 5, Chapter 12, Section 4 for guidance.

So far little mention has been made of the overboard drain from the deck seal and it can be assumed that the materials and arrangements are generally the same as for the scrubber even though the corrosive problems should be much less.

There are however, two points which should be mentioned. The first is that the drain pipe from the deck seal should not be led back into the engine room nor should it join the scrubber effluent overboard line since in both cases there would be a danger of connecting a dangerous area with a safe area, for example, when a drain pipe is removed for repair. It is recommended that, wherever possible, the scrubber effluent drain and deck seal drain should be led overboard through separate cofferdam spaces or ballast tanks. In most cases, however, due to its location, the deck seal drain is led overboard in way of the cargo pump room although, as permitted by paragraph 3, 12, 4.1.8, it can also pass through a cargo tank provided the thickness of the piping is at least the same thickness of the shell plating in way, but need not exceed 19 mm.

1.1.11 Inert Gas Systems on Chemical Tankers and Liquefied Gas Carriers

Part 5, Chapter 15, Section 7, of the Rules deals only with inert gas systems on board ships intended for the carriage of crude oil or oil products having a flash point not exceeding 60°C (closed cup test). It is therefore still necessary to consider inert gas systems on other ships such as chemical tankers and liquefied gas carriers. Complications arise in that whilst part 5, Chapter 15, Section 8 addresses itself to I.G. systems on

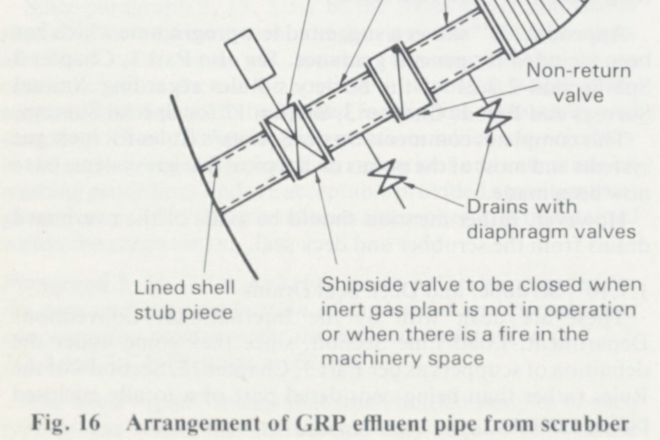


Fig. 16 Arrangement of GRP effluent pipe from scrubber

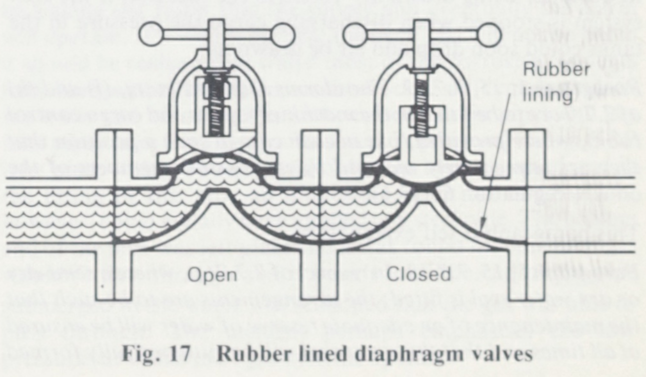


Fig. 17 Rubber lined diaphragm valves

chemical tankers and Chapter IX of the Liquefied Gas Carrier Rules addresses itself to I.G. systems on liquefied gas carriers it is Regulation II-2, 55.5 of the 1981 Amendments to SOLAS 1974 which is, at present, the guiding light throughout the world on these matters. It is hoped to clarify the situation by the following.

1.1.12 Inert Gas on a Chemical Tanker

It is clear that having regard to the wide spectrum of cargoes carried and their differing characteristics that a chemical tanker cannot be regarded in the same manner as a crude oil or product oil tanker and that prevention of fire by using an inert gas system would not be the only consideration. For example, toxicity, corrosivity and reactivity of the cargoes with the inert gas and the danger of polymerisation have to be considered.

The chemical industry, that is the suppliers, carriers, port authorities and storage industries were concerned that by fitting inert gas systems on board chemical tankers the dangers would be enhanced rather than diminished, particularly with regard to the safety of personnel. On a chemical tanker loading and unloading many separate parcels at different ports, personnel may be required to enter cargo tanks at frequent intervals for the purpose of cleaning them to high degrees of cleanliness in order to avoid contamination, or reducing the quality, of the cargo to a point where it was no longer acceptable to the shipper.

Bearing these considerations in mind, the chemical industry was basically not keen to fit inert gas systems on chemical tankers at all. However, the industry recognised that since low flash petroleum based products were already being carried on chemical tankers, these products being the same as those carried on crude and oil product tankers and already covered by Regulation 62, no argument could be put forward for not fitting inert gas for the carriage of these same cargoes except that, having regard to a chemical tanker being a specialised ship, the inert gas system on board would also have to be a specialised system. Therefore as an interim measure and until such time as further consideration could be given to inert gas systems applicable to the carriage of flammable "Chemical" products, as distinct from flammable "Petroleum" products, Resolution A473(XII) was developed and entitled "Interim Regulation for Inert Gas Systems on Chemical Tankers Carrying Petroleum Products". This Resolution can be found on Page 47 of the IMO publication containing the Inert Gas Guidelines.

The purpose then was to apply this Resolution to chemical tankers as equivalent to the application of Regulation 62 to crude and oil product tankers from the dates and for the conditions stipulated in regulation 60 of the 1981 Amendments to SOLAS 1974.

Having satisfactorily dealt with the problem of carrying "Petroleum" products on board chemical tankers, the problem still remained of what to do about the carriage of "Chemical" products also having a flash point below 60°C.

The chemical industry still being concerned about the use of inert gas with chemical products and the possible development of a totally new and expensive concept in inerting especially for those cargoes, set up an Inter-Industry Working Group for the purpose of carrying out an in-depth study of all the aspects concerned with fitting, or not fitting, inert gas systems on board such ships. This working group's approach was to divide the task into two parts the first part being a "qualitative" report and the second a "quantitative" report.

The "qualitative" report took into account an analysis of chemical tankers with respect to their design and operation; the standards of safety employed taking into consideration the requirements for, and characteristics of, the cargoes; the safety records of chemical tankers and the problems associated with the use of inert gas and the comparisons between the operations of a chemical tanker compared with crude oil or product oil tankers.

The "quantitative" report was carried out by Lloyd's Register of Shipping and quantified the risks involved in the following aspects of chemical tanker operations:

(a) the risk of fire and explosions in cargo tanks.
(b) the hazards to crew in entering tanks.
(c) the hazards associated with chemical carry-over from one tank to another via the inert gas main and branch lines.
(d) the dangers of initiating polymerisation of a cargo by use of inert gas.

Since even this second part of the report is about half an inch thick and very comprehensive it obviously cannot be dealt with in any depth in this paper. However, it can be stated that it played its part in the eventual outcome at IMO which was to amend the wording of Regulation II-2, 55.5 by Resolution A566 (14) which is now worded as follows:

Reg. 55.5 as amended
The requirements for inert gas systems of Regulation 60 need NOT be applied to:

1. Chemical tankers constructed before, on or after 1st July 1986, when carrying cargoes described in paragraph 1 (i.e. 55.1), provided that they comply with the requirements for inert gas systems on chemical tankers developed by the Organisation; or*

*2. Chemical tankers constructed before 1st July 1986, when carrying crude oil or petroleum products, provided that they comply with the requirements for inert gas systems on chemical tankers carrying petroleum products, developed by the Organisation**; or*

3. Gas carriers constructed before, on or after 1st July 1986, when carrying cargoes described in paragraph 1 (i.e. 55.1), provided that they are fitted with cargo tank inerting arrangements equivalent to those specified in sub-paragraph 1 and 2, or

4. Chemical tankers and gas carriers when carrying flammable cargoes other than oil or petroleum products such as cargoes listed in Chapters VI and VII of the Code for Construction and Equipment of Ships Carrying Dangerous Chemicals in Bulk (otherwise known as the BCH Code) or Chapters 17 and 18 of the International Code for the Construction and Equipment of Ships Carrying Dangerous Chemicals in Bulk (otherwise known as the IBC Code):

4.1 if constructed before 1st July 1986; or

4.2 if constructed on or after 1st July 1986, provided that the capacity of tanks used for their carriage does not exceed 3,000 m^3 and the individual nozzle capacities of tank washing machines do not exceed 17.5 m^3/h and the total combined throughput from the number of machines in use in a cargo tank at any one time does not exceed 110 m^3/h.

* *Reference is made to Regulation for Inert Gas Systems on Chemical Tankers adopted by the Organisation (i.e IMO) by Resolution A567 (14).*

** *Reference is made to Interim Regulation for Inert Gas Systems on Chemical Tankers Carrying Petroleum Products, adopted by the Organisation (i.e. IMO) by Resolution A473 (XII). "(See page 47 of the Inert Gas Guidelines).*

The above documents make heavy reading and, so far as the Society is concerned, it is hoped to re-phrase this in a more positive form for inclusion in the new Rules for Chemical Tankers to be published in the near future.

In the meantime, however, some interpretations may be of help. Considering item 4.2 first, it seems obvious that having regard to the small volume of cargo tank and small capacity of tank washing machines that these are related to limiting the

generation of static electricity to such an extent that it can no longer be a source of ignition. So, provided the stated parameters are not exceeded, even a 100,000 tons deadweight chemical tanker, whether it be carrying chemical or petroleum based products taken from the Chapters mentioned in item 4, would not need to be fitted with an inert gas system, whereas a crude oil, products or combination oil tanker above 20,000 tons deadweight, would.

Considering item 4.1, this means that an inert gas system is not required on a chemical tanker constructed before 1st July 1986, when carrying flammable chemical products. Such flammable cargoes are those annotated with the letter "F" in column "i" of Chapter VI, or are by definition flammable in Chapter VII, of the BCH Code, or annotated with the letter "F" in column "i" of Chapter 17, or are by definition flammable, in Chapter 18 of the IBC Code.

In contrast, flammable petroleum cargoes are those flammable cargoes included in Appendix 1 of Annex I of MARPOL 1973/78.

Since it is anticipated that most chemical tankers will fall within the confines of items 4.1 and 4.2 previously mentioned, it is not intended to proceed further with interpretations but only to state that should an inert gas system still require to be fitted, because any one of the three parameters mentioned in 4.2 has been exceeded and the tanker is at the same time over 20,000 tons deadweight, then the requirements for the inert gas plant are contained in Resolution A567 (14). It is hoped that the wording of this resolution will, with minor additions and/or amendments, eventually appear in the Society's Rules for Chemical Tankers.

The wording of Resolution A473 (12) and A567 (14) are identical, except for the Preamble and Regulation 1, and also follow very closely the wording of Regulation 62 and, therefore, Also Part 5, Chapter 15, Section 7 of the Rules. However, by virtue of the differing cargo requirements on petroleum and chemical tankers some differences do occur and for instance in Resolutions A473 (12) and A567 (14) these are as follows:

(i) the deck seal can be replaced by an equivalent arrangement such as a double block and bleed valve arrangement (*see* Paragraph 9.1);

(ii) no valve locking arrangements are required;

(iii) there is no requirement to have a cross-connection between the inert gas and cargo mains;

(iv) no pressure vacuum breaker is required.

These differences arise because, whereas on crude oil and product tankers the inert gas lines are permanently in communication with the cargo tanks, on chemical tankers flammable cargoes requiring inerting may be carried at the same time as chemical cargoes which may not require inerting, e.g., sulphuric acid. In such a case, if the cargo tanks were connected through the inert gas branch lines to a common main, there could be a danger of one or more cargoes reacting unfavourably with another either hazarding personnel or affecting the quality of the cargoes concerned. It is basically for these reasons that on chemical tankers the inert gas lines can be disconnected for those cargoes not requiring to be inerted. In other words, both as regards inerting and venting on chemical tankers each cargo tank tends to be treated individually, especially where cargo tanks are fitted with individual shaft driven deep well or submerged hydraulically operated pumps.

Now that Resolution A566 (14) has defined the applications of inert gas to chemical and liquefied gas carriers and Resolution A567 (14) has described the inert gas system to be fitted; what remains? The quality of the inert gas remains because it is still of importance to the shipper to avoid contamination, polymerisation or other reactions and inert gas derived from oil fired equipment such as a boiler or inert gas generator may not be suitable unless some of the reactive elements are removed.

In the combustion process oil fuel and air are burned and convert to carbon dioxide, plus water, plus nitrogen. If the carbon dioxide and water can be removed only nitrogen will be left and this, being completely inert, is ideal for use with sensitive cargoes.

If the inert gas is required for use with water reactive chemicals or on low temperature liquefied gas carriers, where the water could turn to ice and block pipes, valves and control systems, the water can be removed by separation and drying processes and the carbon dioxide by absorption in chemical sieves thus leaving almost pure nitrogen which can be led direct to the cargo tanks or hold spaces or compressed and stored under pressure on board.

Figure 18 shows a typical inert gas system where the inert gas is first generated from an inert gas generator, passed through molecular sieves where the carbon dioxide and water are removed and the remaining nitrogen gas finally compressed and stored in a deck tank for use in the cargo tanks as necessary.

Alternatively nitrogen in its liquid form, can be loaded direct into "vacuum flask" type storage tanks on deck and the nitrogen inert gas then produced by passing the liquid through vaporizers as required (*see* Fig. 19).

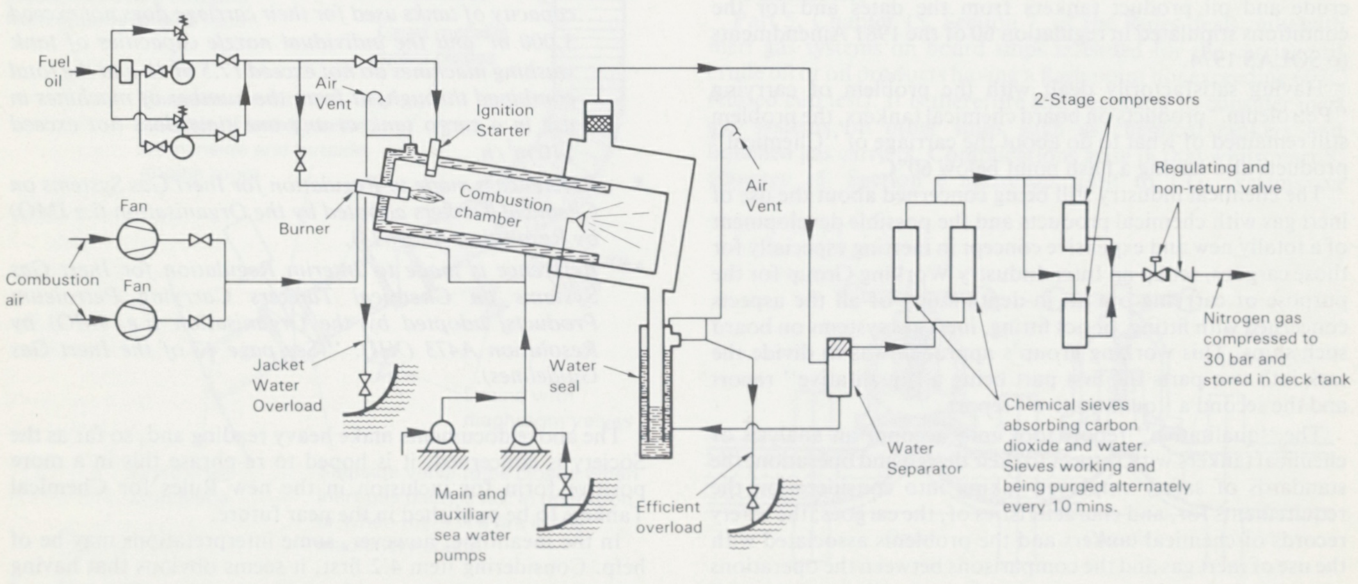


Fig. 18 Nitrogen gas developed from I.G. generator plant

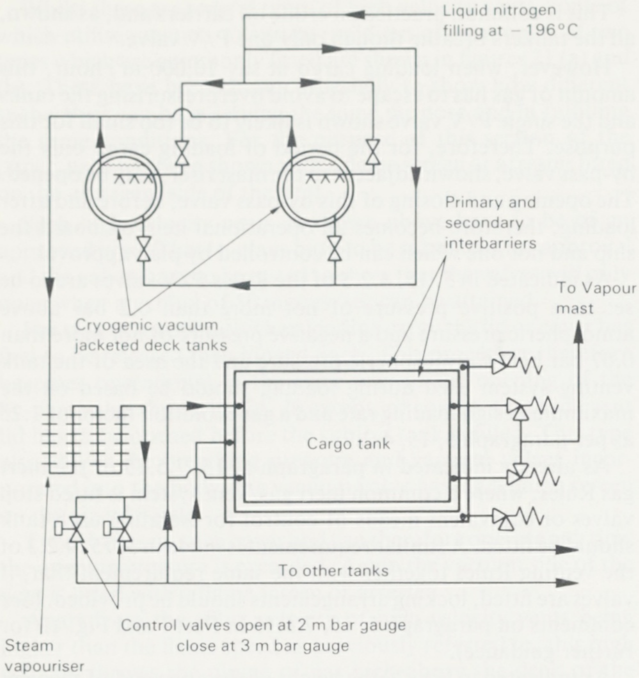


Fig. 19 Nitrogen gas inerting of interbarrier spaces on a liquefied natural gas (LNG) Carrier.

There are other systems which are also capable of producing nitrogen gas on board ship such as pressure swing absorption and membrane or separation systems but these appear to have had only restricted or specialised usage in the marine field and are mentioned only in passing.

It is understood that there are some chemical cargoes where even pure nitrogen is unsuitable for inerting. This, for example, is where certain monomers require the presence of oxygen to inhibit or prevent polymerisation and therefore such monomers provide the exception that can be found for any rules and would have to be dealt with on a separate basis and probably in accordance with operational guidelines developed by the chemical industry itself.

1.1.13 Inert Gas on a Liquefied Gas Carrier

An inert gas system on a liquefied gas carrier will also follow Resolution A567 (14) but at the present time it is not clear how the requirements are to be incorporated into Chapter 9 of the Liquefied Gas Ship Rules although no great difficulties are envisaged.

Obviously the low temperatures on a gas ship could not tolerate the presence of water in a deck seal and a double block and bleed arrangement would be the most likely equivalent arrangement.

Nevertheless, deck water seals or equivalent arrangements are fitted when a gas ship is carrying naphtha or other low boiling cargo. In such a case the deck seal is incorporated into the inert gas system during carriage of the naphtha and then by-passed again when the ship reverts to the carriage of liquefied gases. It should also be mentioned that during the carriage of such non-boiling cargoes the vent mast outlet should be protected by a flame screen as indicated in Paragraph 17.15 of the gas ship Rules.

1.1.14 Conclusions on Part I

It is perhaps now safe to say that any further developments on the means of generating inert gas and its use in the marine industry are likely to be few and far between. The grey period from the initiation and through the evolution and development of new or amended rules and regulations appear to have come to rest in the latest publications referred to in this paper.

As a final comment it should be stressed that any inert gas system is only as good as its maintenance. In the same way that it produces an inert gas it is also an inert installation in that there are no obvious moving parts to draw attention to its need for maintenance which is vital for the plant to perform its primary function, which is to prevent the formation of flammable atmospheres within cargo tanks.

As has been stated elsewhere "Safety does not happen. it is the reward of care, thought and good organisation".

Glossary of Terms

Inert Gas means a gas or a mixture of gases, such as flue gas, containing insufficient oxygen to support the combustion of hydrocarbons.

Inert condition means a condition in which the oxygen content throughout the atmosphere of a tank has been reduced to 8 per cent or less by volume by addition of an inert gas.

Inert gas plant means all equipment specially fitted to supply, cool, clean, pressurise, monitor and control delivery of inert gas to cargo tank systems.

Inert gas distribution system means all piping, valves, and associated fittings to distribute inert gas from the inert gas plant to cargo tanks, to vent gases to atmosphere and to protect tanks against excessive pressure or vacuum.

Inert gas system means an inert gas plant and inert gas distribution system together with means for preventing back-flow of cargo gases to the machinery spaces, fixed and portable measuring instruments and control devices.

Inerting means the introduction of inert gas into a tank with the object of attaining the inert condition.

Gas freeing means the introduction of fresh air into a tank with the object of removing toxic, flammable and inert gases and increasing the oxygen content to 21 per cent by volume.

Purging means the introduction of inert gas into a tank already in the inert condition with the object of:

(i) further reducing the existing oxygen content; and/or

(ii) reducing the existing hydrocarbon gas content to a level below which combustion cannot be supported if air is subsequently introduced into the tank.

Topping-up means the introduction of inert gas into a tank which is already in the inert condition with the object of raising the tank pressure to prevent any ingress of air.

Flammable refers to any liquid having a flashpoint not exceeding 60°C (closed cup test) as determined by an approved flashpoint apparatus and whose Reid vapour pressure is below atmospheric pressure.

PART II VENTING SYSTEMS

HISTORY

The history of venting is as old as the history of tanks in that if no vents are provided the tanks will rupture due to over pressure when being loaded and will again rupture due to vacuum when being unloaded.

2.1 General Comments

The term venting in the context of this paper refers, in the main, to the expulsion of air, inert gas and other gases, or combinations of these from within the cargo tanks of crude oil, product oil and chemical tankers and liquefied gas carriers and includes the processes of purging and gas freeing. It also includes the inflow of air, inert gas and other gases under vacuum conditions. It does not include ventilation systems such as these fitted for the ventilation of cargo and ballast pump rooms and other spaces which require ventilation for compliance with Rule requirements, although brief mention will be made of such systems later on.

Naturally the means and modes of venting can vary to some extent depending on the type of ship and tanks still rupture, not from the fact that someone has forgotten to fit vents or air pipes, but because the vents are too small for the large capacity pumps fitted; or they are fitted with flame screens or flame arresters which have become blocked; or due to the design of some fancy vent heads that can be fitted, changes of direction of flow cause a higher resistance than normal which has not been taken into account.

2.1.1 Classification Rules

The Society's Rules for venting are contained in Part 5, Chapter 15, Section 4 and, in particular, paragraph 5, 15, 4.1.3 states that "Cargo tank venting arrangements are to be designed to provide:

(a) pressure/vacuum release of small volumes of vapour/air mixtures flowing during a normal voyage and,

(b) venting of large volumes of vapour/air mixtures during cargo handling and gas freeing operations."

Considering (a) above, this relates to the tanks being able to breathe. In other words this takes into account the effects of temperature variations such that during a hot day the gases in the tanks expand and may escape through the pressure side of the P/V (pressure/vacuum) valve and during the cold night the gases contract and air may be drawn in through the vacuum side of the P/V valve. The operative word in the foregoing sentence is "may" because P/V valves are set to certain pressure and vacuum conditions, say, 1400 mm water gauge on the pressure side and 350 mm water gauge below atmosphere on the vacuum side and, taking into account the compressibility of the gases within the tank, the temperature variations may not be large enough to cause the P/V valves to lift. Any influx of air could of course influence whether or not the atmosphere in the tank becomes flammable but would usually have little effect in a full tank since the atmosphere is likely to be over-rich anyway and the effect on any empty tank is relative to the small amount of air which will enter compared to the large capacity of the empty cargo tank.

It will perhaps be obvious by the wording of (a) that any release or inflow of vapour/air mixture should be through the P/V valves and not through open vents and account should be taken that where an inert gas system is fitted the vapour/air mixtures mentioned in (a) and (b) would also include inert gas.

2.1.2. Contents

Figure 20 shows a typical venting system on a crude oil tanker where the inert gas main and branch lines have been utilised to form a venting system which is common to all cargo tanks.

This is common practice on crude oil carriers and, as shown, all the tankers breathe though only one P/V valve.

However, when loading cargo at say 10,000 m^3/hour, this amount of gas has to escape to avoid overpressurising the tanks and the single P/V valve shown is likely to be too small for this purpose. Therefore, for the period of loading cargo only the by-pass valve, shown adjacent to the mast riser, may be opened. The opening and closing of this by-pass valve, before and after loading, therefore becomes an operational item on board the ship and not one which can be controlled by plan approval.

As indicated in 5, 15, 4.2.5 of the Rules P/V valves are to be set to a positive pressure of not more than 0.2 bar above atmospheric pressure and a negative pressure of not more than 0.07 bar below atmospheric pressure and the area of the tank venting system used during loading should be based on the maximum design loading rate and a gas evolution factor of 1.25 as per paragraph 5, 15, 4.2.7.

As already indicated in paragraph 5, 15, 7.5.15 of the inert gas Rules, where a common inert gas/vent system is fitted stop valves or equivalent means of control for isolating each tank should be fitted. A similar requirement is made in 5, 15, 4.2.3 of the venting Rules together with the same requirement that, if valves are fitted, locking arrangements should be provided. (*See* comments on paragraph 5, 15, 7.5.15 in Part I and Fig. 13 for further guidance).

As indicated in Fig. 20 the height of the open end of the vent mast for discharging free flowing vapours during loading, that is when the by-pass valve is open, should be 6 m above the deck or above the fore and aft gangway on a tanker if fitted within 4 m of the gangway and the open end should be designed so that the vapours are directed in an upward vertical direction. The open end should also be provided with a device to prevent the passage of flame, such as a flame screen or flame arrester, but reference to this is made later.

2.1.3 High Velocity Venting System

As an alternative to having a vent system common to all cargo tanks it is quite common these days to have a high velocity vent fitted to each tank. High velocity vents are designed in such a way that they will open only when the pressure within the tank is raised to such an extent that, on opening, the efflux velocity of the gases will immediately be not less than 30 metres/sec and that this velocity will not drop below 30 m/sec even when the valve is closing. This velocity is estimated to be twice the speed at which any flame front burning through an unconfined cloud of gas external to the tank could approach the opening of the vent or, in other words, the flame front should never be able to get near enough to the vent to be able to flash back into the cargo tank.

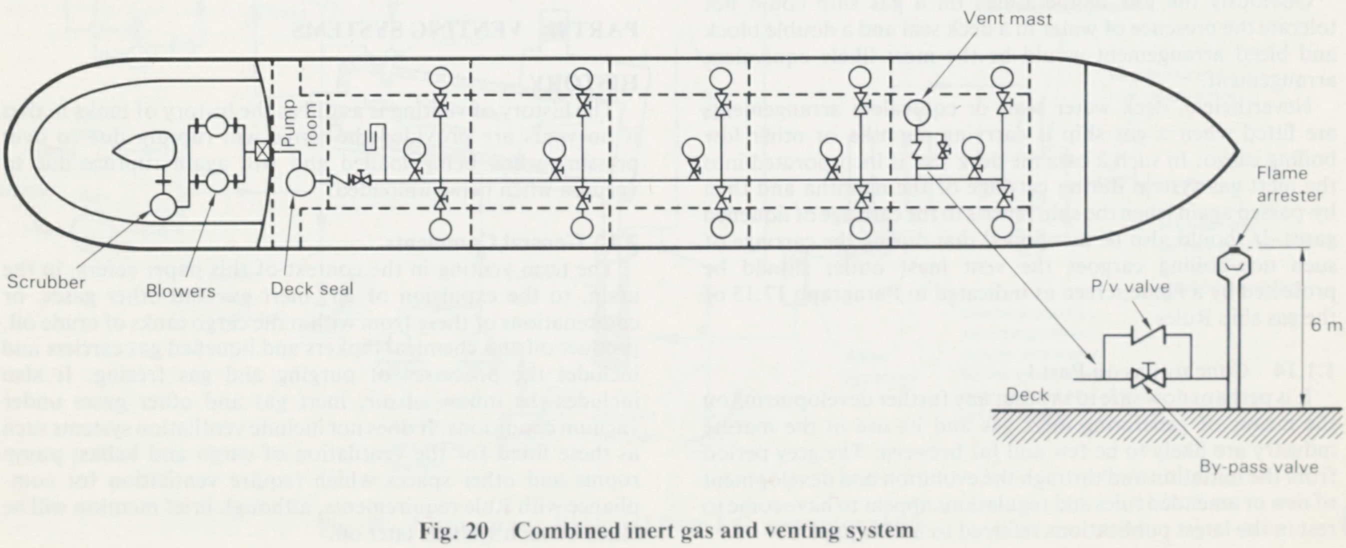


Fig. 20 Combined inert gas and venting system

Whilst there are several types of high velocity vents, some of which utilise magnets to ensure rapid cut off, examples of two types which are commonly fitted are shown in figures 21 (a) and (b). These have been chosen especially to show how different such valves can be in serving the same purpose and in fulfilling the same conditions. For the purpose of this section of the paper, each has been shown with a flame screen or arrester fitted on the upstream side of the vent.

Such high velocity vents as shown above have to be of an approved type. That is, plans have to be submitted for approval and the valves have to be tested to show that the valves will only open when an efflux of 30 metres/sec can be attained.

For type (a) to operate when loading cargo the lid or cover has first to be released then hinged back. This procedure therefore becomes operational in the same way as the opening of the by-pass valve shown in Fig. 20 except, of course, that each vent lid has to be opened before the subject tank is filled. This type also shows the breathing pressure and vacuum valves incorporated into the body, the vacuum valve having a flame screen on the air-inlet side.

Type (b) has no such cover and can therefore operate any time the opening pressure is exceeded. Again the vacuum side of the vent is fitted with a flame screen or arrester.

Apart from the efflux velocity of high velocity vents being higher than the flame speed as previously referred to, this high velocity throws the plume of gas high above the deck of the tanker and at the same time entrains the surrounding air so diluting the gas and thereby reducing the concentration of the hydrocarbon content to a safe or "too lean" condition.

2.1.4 Heights and Location of Vents on Oil and Chemical Tankers

In the International Safety Guide for Oil Tankers and Terminals (ISGOTT) sketches will be found of experiments carried out on board tankers under sea going conditions showing how the plumes of gases from the vents are affected by the direction and speed of ships under various wind conditions and how these gases can be trapped in various places on deck, some of which would normally be considered to be safe. It is experiments like these which helped to frame the requirements for the heights and locations of vent outlets from mast risers and high velocity vents.

As indicated in paragraph 5, 15, 4.5.3 the height of vents fitted with high velocity vents and also for pressure vacuum valves which are used for breathing purposes only may be reduced to 2 metres above the deck, or 2 metres above the fore and aft gangway if located within 4 metres of the gangway. Should the breathing P/V valve be separate from the high velocity vent it can be located near to the gangway but obviously not underneath it.

For both the vent masts and high velocity vents the distance between the outlets and the nearest openings to accomodation and enclosed working spaces, or to sources of ignition, should be not less than 10 metres. In the case of breathing P/V valves this distance can be reduced to 5 metres as indicated in 5, 15, 4.5.1. Normally it is expected that these distances are measured horizontally since if this were not so it could mean a 12 metre high vent immediately above a 2 metre high source of ignition. This is certainly not the intention since most gases are heavier than air and the existing gas would drop straight down on to the source of ignition.

To interpose a short story from several years ago, IACS, the International Association of Classification Societies, used to have formulae and graphs for establishing the vertical and horizontal locations of vent outlets and on one occasion a colleague in Sydney quite correctly, estimated that the height of vent should be 30 metres and what should he do about the Sydney Bridge? The answer could well have been to convert the bridge into the bascule type like the Tower Bridge to make the ex-patriots feel more at home. However, common sense had to prevail and if a comparision can be made between that 30 metres

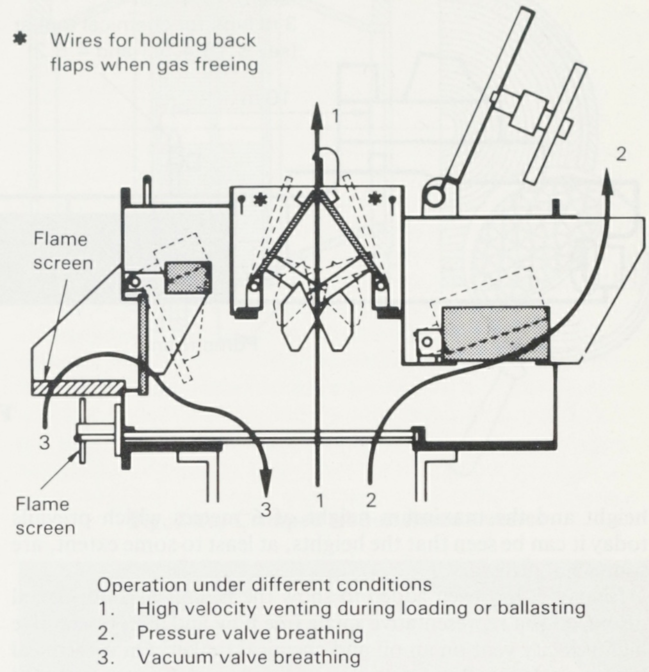


Fig. 21(a) Example of high velocity vent

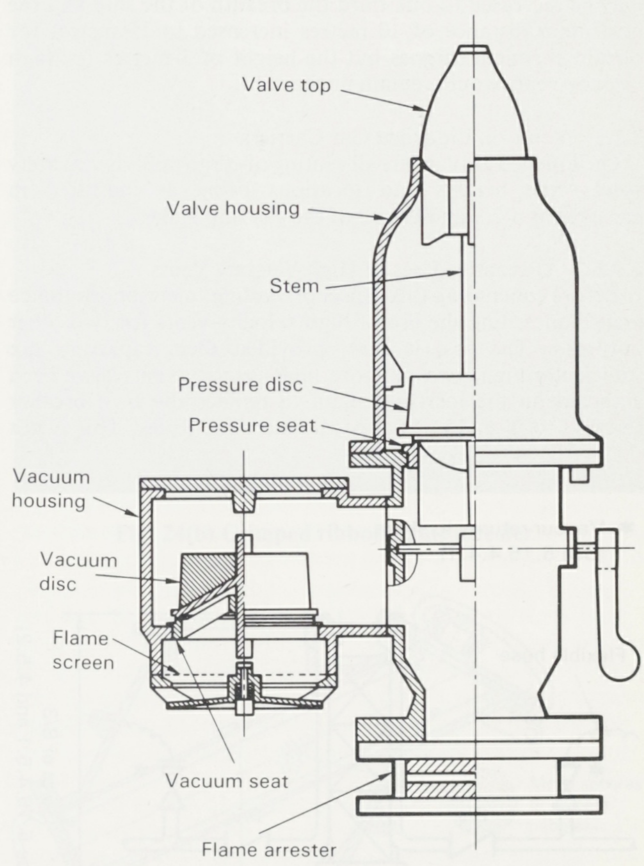


Fig. 21(b) Example of high velocity vent

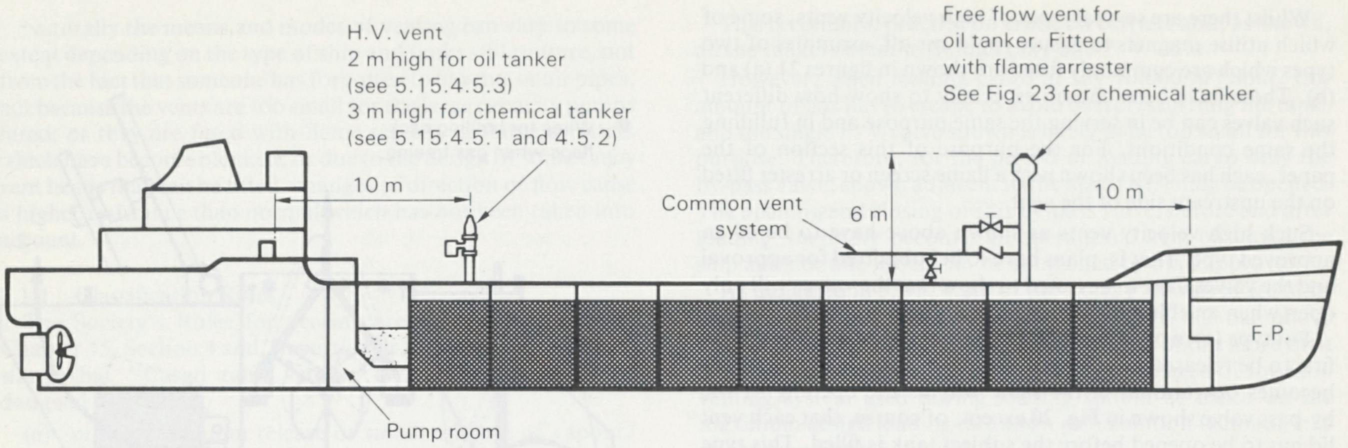


Fig. 22

height and the maximum height of 6 metres which prevails today it can be seen that the heights, at least to some extent, are somewhat arbitrary.

Figure 22 has been added to show the heights and horizontal distances of a representative single free flow and a representative high velocity vent on an oil and chemical tanker. On a chemical tanker single free flow vents may be fitted to each cargo tank and bunched together to form a structured mast as shown in Fig. 23 where usually no by-pass valve is fitted since each P/V valve has sufficient capacity to cope with the smaller rate of loading.

As indicated in paragraph 5, 15, 4.5.2 the height of 6 metres may be increased to one-third the breadth of the ship and the horizontal distance of 10 metres increased to 15 metres for certain chemical cargoes but the height of 3 metres for high velocity vents would remain unchanged.

2.1.5 Vents on Liquefied Gas Carriers

On liquefied gas carriers all venting of cargo tanks is via safety valves, the heights and locations being as indicated in paragraphs 8.2.9 and 8.2.10 of the gas ship Rules.

2.1.6. Alternative Uses of High Velocity Vents

Before concluding this aspect of venting, mention should be made concerning the use of high velocity vents for two other purposes. The first is that, provided their capacities are sufficiently high, two or more high velocity vents have been accepted on the inert gas main to replace the P/V breaker referred to in 5, 15, 7.6.3 of the inert gas Rules. This is not

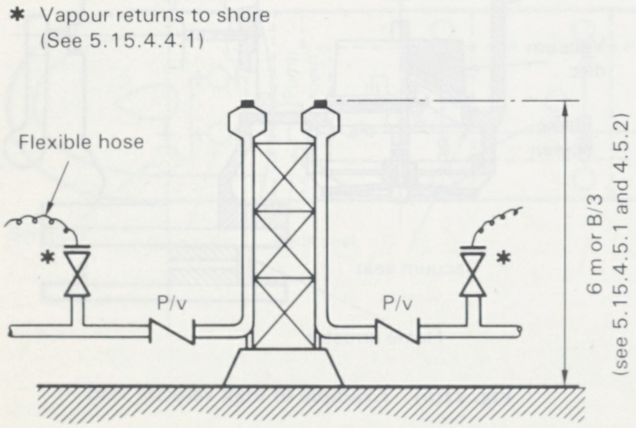


Fig. 23 Vent mast for chemical tanker

common practice and more than one high velocity vent would be required to ensure that one or other of them will still operate despite any sticking from hydrocarbon deposits.

The second is the use of a single high velocity vent on the inert gas or common vent main with an isolating valve located below it. The sole purpose of this valve, if fitted, is to rapidly relieve any pressures in the cargo tanks so that any necessary manual operations such as sounding or sampling the tanks through openings can be carried out. This valve is therefore set to open at a very low operating pressure, say 170 mm water gauge, and, since it is normally isolated, it does not form part of the venting system.

2.1.7. Devices to Prevent the Passage of Flames

Now comes the difficult part and it stems from a rather innocuous phrase in Regulation II-2, 59.1.5 of the 1981 Amendments which states that "The venting system shall be provided with devices to prevent the passage of flame into cargo tanks. The design, testing and locating of these devices shall comply with the requirements established by the Administration which shall contain at least the standards adopted by the Organisation."

To explain this paragraph a little further the "Administration" for British Ships would be the Department of Transport (DoT) and in the U.S.A. the United States Coast Guard (USCG) and so on, whilst the "Organisation" is IMO and the "standards" are those given in IMO document MSC/CIRC.373. "MSC" is the Maritime Safety Committee at IMO which deals with technical matters and meets twice a year.

Any Administration could, if it wished, include additional requirements to those stated in MSC/CIRC.373.

If it is explained that these "standards" took three separate weeks at IMO over an 18 month period to draft with prior meetings at the International Chamber of Shipping (ICS) and no doubt many meetings by individual Administrations and classification societies, testing laboratories and other bodies on a world wide basis it can be seen that the ratio of expansion in the workload from the wording of the original paragraph to the final product is immense. It is perhaps sufficient to say that the subject of "devices to prevent the passage of flame" is of great interest and very much a matter for experts of which there seems to be comparatively few. Even the experts themselves, in excellent papers available on this subject, admit that there are gaps in their knowledge and the Author is the first to admit that his gap is such that even a small flame could pass through it.

However, what is a "device" to prevent the passage of flame"? By definition these can be stated as:

"Flame screens"—are devices utilising wire mesh to prevent the passage of unconfined flames, that is flames in the open air which are not contained with pipework or tanks.

"Flame arresters" or "detonation arresters"—are devices utilising, in most marine cases, crimped metallic ribbon where the flame arresting element is based on the principle of quenching.

"High velocity vents"—which consist of a mechanical valve which adjusts the opening available for flow in accordance with the pressure at the inlet of the valve in such a way that the efflux velocity at the outlet cannot be less than 30 metres/second.

"Inert gas systems"—some controversy arose over this last item, which will be dealt with later.

Taking the above in turn:

2.1.7.1 Flame Screens

In the Author's opinion the best definition of a flame screen is that found in the Coast Guard, Department of Transportation, Code of Federal Regulation 46, Parts 30 to 40, Paragraph 30.10-25 which states:

"The term 'flame screen' means a fitted single screen of corrosion resistant wire of at least 30 × 30 mesh, or two fitted screens, both of corrosion resistant wire of at least 20 × 20 mesh, spaced not less than half inch or more than one and a half inches apart."

Remembering that the USA still adhere to inches rather than centimetres this means 30 × 30 per inch equals 900 meshes per square inch (equals 12 × 12 per cm = $144/cm^2$) and 20 × 20 per inch equals 400 meshes per square inch (equals 8 × 8 per cm = $64/cm^2$). The important item which is missing from the above is the diameter of wire because if the diameter is too large the gaps will be filled in to form a solid plate. The diameter of wire for the 30 × 30 mesh is 0.012 inches (0.3 mm) and for the 20 × 20 mesh 0.016 inches (0.4 mm). In any event the clear area through flame screens or arresters should be at least 1.5 times the cross-sectional area of the vent line.

The single gauze may be better than the double gauze under test conditions since the double gauze could absorb and retain a greater amount of heat and cause flashback more readily.

In practice there is some doubt that a flame screen, whilst efficient in preventing the flashback of unconfined flames into a cargo tank, would not be efficient in preventing the passage of flame under sustained burning or, in other words, would not survive the endurance burning test as referred to in paragraph 3.2.3 of MSC/CIRC.373.

It is for this reason that flame screens were originally only allowed to be fitted (a) on the vacuum side of P/V valves and (b) at the inlets within the cargo area which were being used for gas freeing, for example at "butterworth" openings when using portable gas freeing fans. As indicated in Part 1 in the comments to 5, 15, 7.2.3 (d) there is still some discussion going on at IMO regarding this subject.

Having stated the limitations in (a) and (b) above, subsequent consideration at IMO of an inert gas system as a device to prevent the passage of flame allowed that for ships fitted with an inert gas system the "devices" would not need to pass the "endurance burning" test and therefore a flame screen could be accepted where, without inert gas, a "flame arrester" would normally be required.

2.1.7.2 Flame Arresters

There are several types of flame arrester such as crimped ribbon arresters, parallel plate arresters, pebble arresters, hydraulic arresters, wire packed arresters, packed tower arresters and sintered arresters. Each type has its own characteristics and they are not necessarily all suitable for the same services. For example, packed tower and pebble arresters are heavy; parallel plate arresters are both heavy and expensive and all these types are mainly utilised in the exhausts from diesel engines of trucks designed for operation in hazardous areas. Figs. 24 and 25 have been included to give a visual impression of the internals of two such arresters.

Since crimped metal arresters are most commonly fitted on board tankers this is the type that will be considered in detail.

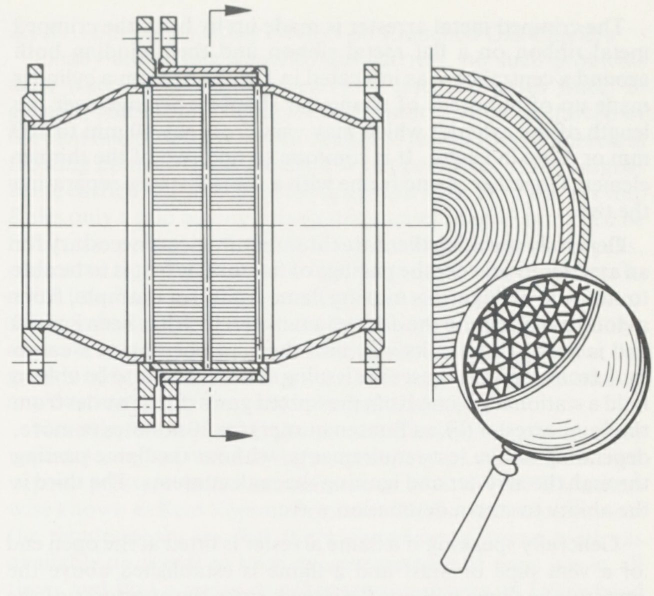


Fig. 24(a) Crimped ribbon flame arrester

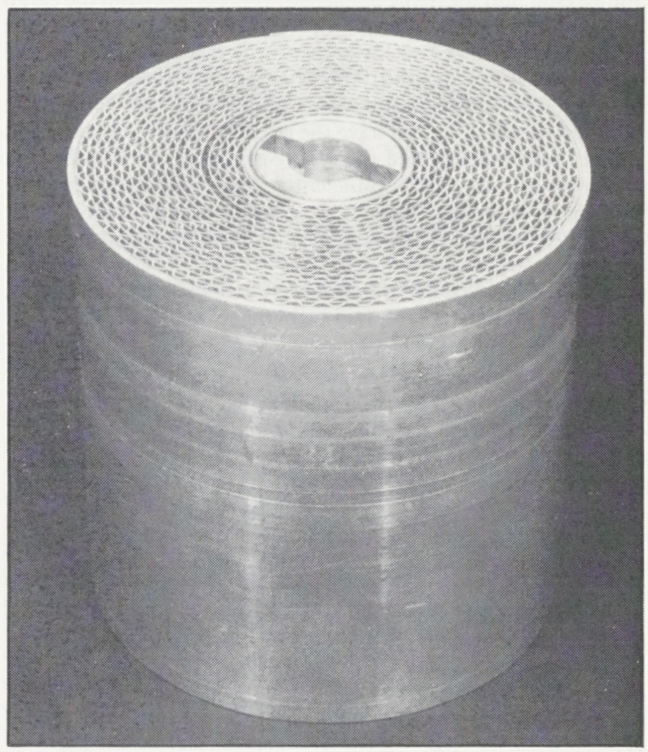


Fig. 24(b) Crimped ribbon flame arrester

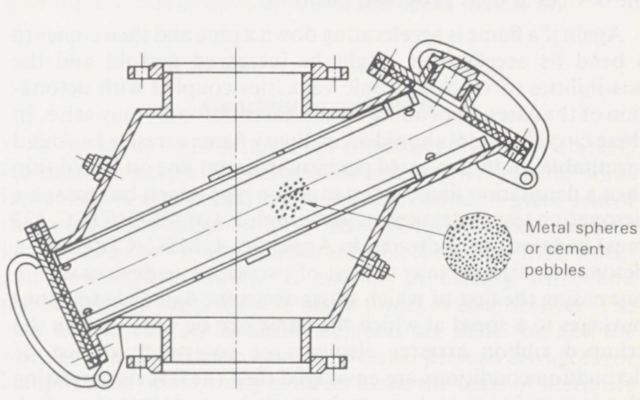


Fig. 25 Granular packed flame arrester

The crimped metal arrester is made up by lying the crimped metal ribbon on a flat metal ribbon and then winding both around a central core as indicated in Fig. 24 to form a cylinder made up of hundreds of triangular shaped openings over the length of the cylinder which may vary from say 10 mm to 100 mm or more in length. It is common to find two of the thinner elements together in one frame with a short distance separating the two.

Generally speaking there are three requirements necessary for an arrester to prevent the passage of flame. The first is to be able to stop and extinguish a moving flame front, for example, from a cloud of vapour on the deck of a tanker which has been ignited and is trying to burn its way into the vent opening of a cargo tank from which the gases are issuing. The second is to be able to hold a stationary flame from pre-mixed gases on the outlet from the flame arrester (like a Bunsen burner) for 30 minutes or more, depending on the test requirements, without the flame passing through the arrester and igniting the tank contents. The third is the ability to stop a detonation wave.

Generally speaking if a flame arrester is fitted at the open end of a vent pipe or mast and a flame is established above the arrester the flame will not flash back since the apertures of the arrester are designed such that the flame is cooled in the mass of the arrester and the flame quenched. If the efflux from the cargo tank is fast enough, for instance from a high velocity vent, the flame could be lifted so far above the arrester that it could be blown out. However, whilst the flame is in close proximity to the arrester there is the question of radiation from the flame to be considered since this will have the effect of heating up the mass of the flame arrester and the casing in which it is housed to a temperature which, if sufficiently high, could exceed the auto-ignition temperatures of the gases and cause flash back. It is to test the radiation aspect that the "endurance burning" test referred to in paragraph 3.2.3 of MSC/CIRC.373 is carried out. Should the open end of the vent have a cowl or head attached, then these attachments should be in place during the test.

If instead of putting the flame arrester at the open end of a vent mast it is placed some distance down the pipe, for example to make it more accessible for cleaning, then a flame from outside and entering the pipe would be accelerated within the confines of the pipe with the chances that if the flame approaches and enters the flame arrester at too high a speed the flame will pass through the arrester before it has had time to be cooled and quenched to such an extent that it is no longer a source of ignition. So again, if it is intended to fit a flame arrester at any position other than at the open end, the test of that flame arrester should be conducted with the relevant length of pipe downstream attached. At one time in the discussion at IMO it was proposed that any arrester fitted within 100 diameters of the open end could be regarded as though the arrester were fitted at the open end, that is, the arrester could be tested by itself. However, since different testing laboratories use different criteria it was thought best to be safe rather than sorry and test the devices at their proposed positions.

Again if a flame is accelerating down a pipe and then comes to a bend its acceleration might be increased tenfold and the possibilities of reaching sonic velocities coupled with detonation of the gases, depending on certain conditions, may arise. In these circumstances should an ordinary flame arrester be found unsuitable for its proposed position in a vent line on board ship then a detonation flame arrester of the type which has passed a detonation test as detailed in paragraph 3.4 of MSC/CIRC.373 must be fitted. As indicated in Appendix 4, MSC/CIRC.373 a detonation arrester may consist of two separate devices within one casing the first of which slows down the flame via tortuous passages to a speed at which the flame can be stabilised on the crimped ribbon arrester elements or equivalent device. If detonation conditions are envisaged then the test rig, arresting elements and pipework must obviously be made strong enough to withstand the shock loadings.

2.1.7.3. High Velocity Vents

The action, heights and location of H.V. vents has already been discussed but in reading MSC/CIRC.373, paragraph 2.2.11 states that "means shall be provided to check that any valve lifts easily without remaining in the open position" and it is perhaps obvious that if a H.V. vent is able to be held open and the opening is greater than that which would prevent flashback then flashback could occur in the presence of a flammable atmosphere and a source of ignition.

Furthermore, paragraph 2.2.12 of MSC/CIRC.373 gives the impression that H.V. vents are not equipped with flame screens or flame arresters. However, as seen from Figures 21 (a) and 21 (b) both types of H.V. vents are shown to have such devices fitted. In Fig. 21 (a) the device is a flame screen of two separate meshes about 10 mm apart and in Fig 21 (b) the housing below the vent contains a flame arrester of crimped ribbon. However, no mention is specifically made of the location of these flame screens and flame arresters in the testing processes for high velocity vents referred to under paragraph 3.3 of MSC/CIRC.373.

It is important to realise that provided H.V. vents pass the flashback and endurance burning tests described and provided that the vents cannot be held permanently open, no flame screens or flame arresters are required. It may be of interest to know that the omission of flame screens in H.V. vents was first considered by IACS some years ago as indicated in paragraph 6 of Recommendation No. 3 of the IACS Book. The reason for wishing to omit the flame screens was that if the flame screen became blocked the strength of the blocked flame screen relative to the small area of vent pipe was inherently much greater than the large diaphragm plates used in the construction of the cargo tanks with the result that the tank would rupture first.

However, the H.V. vent shown in Fig. 21 (a) can be held permanently in the open position by means of the wires shown on the sketch which loop over the hinged flaps in the "held open" position when gas freeing or ballasting the cargo tank and a flame screen or arrester would therefore always be required unless, of course, the wires were dispensed with. The H.V. vent shown in Fig. 21 (b) can also be held open by a cam operated by the handle at side but the handle is so designed that it will fall as soon as it is released and the valve will close. Therefore, provided the tests have been passed, the flame arrester could be dispensed with.

Before considering item 2.1.7.4, the inert gas system, reference should be made to some aspects of MSC/CIRC.373 which perhaps require particular emphasis.

Firstly the flashback, endurance burning and detonation tests referred to are intended to apply only to the aliphatic and aromatic group of hydrocarbons. Should the devices be intended for use with more flammable and volatile liquids and/or gases then the tests would require to be carried out using those gases. For example carbon disulphide is flammable in the range of 1.3 to 44 percentage volume in air and ethyl ether in the range of 1.7 to 48%, both of which are far higher than the flammable range of propane or gasoline which are the usual testing mediums as indicated in paragraph 3.1.3. However, a subtle point in 3.2.3.2 indicates that only gasoline may be used for the "endurance burning" test and this is because gasoline has a lower auto-ignition temperature than propane and in the "endurance burning" test, which is a heat radiation test, flashback is therefore more likely to occur.

Finally, as regards tests, it is of interest to note in 1.2.7 that ignition by lightning has not been considered since insufficient information is available to simulate the conditions of a lightning strike.

2.1.7.4. Inert Gas System

If the subject of this section is "devices to prevent the passage of flame" then an inert gas system is not a device—it is a system—but it cannot be denied that, by virtue of rendering the atmosphere within a cargo tank to a non-flammable condition

whereby a flame front trying to enter the tank would be extinguished, it does have a similar end result. However, at IMO, a division of opinion arose on this matter. There was one camp which considered that an inert gas system should be considered as a device to prevent the passage of flame which would eliminate the need for fitting flame screens or arresters. There was also another camp which said "not so", because an inert gas system can break down and then the flame prevention capability would be lost. The counter claim to this by the first camp was that even if the inert gas system broke down the Inert Gas Guidelines details the precautionary measures to be taken before discharging a cargo. So, as ever, since the two camps were evenly divided a compromise had to be reached. This was, that for ships fitted with inert gas systems, flame screens, flame arresters and high velocity vents would require to pass only the flashback tests and not also the endurance burning test.

In the Author's opinion the term "device" conjures up a vision of a self contained object capable of being tested to the full requirements in a test rig whereas an inert gas system is composed of many parts dependent to some extent on the other and the failure of one or more parts could cause the system to be put out of effective action. It is therefore considered that an inert gas system is not a "device" in the true sense of the word and, that whatever emergency procedures are adopted in the event of inert gas breakdown, the conditions cannot be as safe as when flame screens or arresters are required to be fitted.

Indeed, the emergency procedures referred to in paragraph 8.3.1 and 8.3.2 of the Guidelines state that, in the event of a breakdown of the inert gas system, the flame arresting devices must be fitted. It is questionable if in such circumstances the required number, size and type of flame arresting devices would be readily available and if it would be better to have fitted them in the first place. It is appreciated of course that the maintenance of flame screens and arresters is a big job and perhaps not always easily carried out in a sea going environment, especially when they may be located at the top of a mast and be heavy to manhandle. However, with proper supervision and execution it should be possible to set up a suitable maintenance programme.

A final thought on those devices which have been tested for flashback only is what distinguishes one which has passed "flashback" only and one which has passed both "flashback" and "endurance burning"? Apart from a small nameplate—probably nothing. It is therefore of great importance that those devices which have passed the flashback test only should be fitted only on ships which are fitted with an operative inert gas system.

2.1.8 Flame Arresting Devices on Chemical Carriers

Having exhausted the design and testing features of these devices the problem of chemical tankers again rears it head in a similar manner to that of the inert gas system on oil and chemical tankers.

The chemical industry are at present putting a case forward, in the Author's opinion a logical one, that in certain circumstances flame arresting devices should not be fitted to the vents of chemical tankers because it may be more dangerous to fit them than to omit them.

The circumstances referred to are the following:

(i) Where cargoes can polymerise and block up the vents.

(ii) Where cargoes have a melting point above 0°C and can solidify if not heated.

(iii) Cargoes which cause increased corrosion.

(iv) Cargoes which are so volatile that special precautions have to be taken.

Another viewpoint is that individual vents on a chemical tanker, each fitted with a P/V valve but with no additional by-pass valve, do not need to be provided with a flame arresting device. However, these proposals are not yet finalised.

2.1.9. Flame Arresting Devices on Liquefied Gas Carriers

Finally addressing liquefied gas carriers, no such problems arise when these ships are carrying liquefied gases since, in general, flame screens or arresters cannot be fitted at the top of the vent masts because of the danger of the meshes or apertures freezing up from the normally low temperatures of the cargoes being carried. As indicated in paragraph 12.1.11 of the gas ship Rules only a grid having a mesh not greater than 13 mm need be fitted, the purpose of which is no doubt to prevent seagulls, overcome by the gases, from dropping down and blocking the relief line from the cargo tank safety valves. Certain types of safety valves could open to an extent which would exceed the aperture necessary to quench a flame, should one occur, but the gases which would be emerging under such conditions are, in any case, generally too rich to be flammable.

Nevertheless, liquefield gas carriers also, on occasions, carry naphtha or other non-boiling cargoes, that is, cargoes having a vapour pressure below atmosphere at 37.8°C (100°F), otherwise known as Reid Vapour Pressure. On such occasions two of the requirements are that there should be sufficient inert gas available on board to keep an overpressure of inert gas at all times and that a gauze or other approved type of safety head should be fitted at the top of the vent mast instead of the grid previously mentioned. Because of the subsequent dangers of icing up it is important that the gauze be replaced with the grid when the ship reverts to the carriage of liquefied gases. Since these two requirements are indicative of oil tanker practice a final requirement is that the inert gas should either pass through a deck seal, which can be by-passed in normal service, or alternatively, pass through a double block and bleed arrangement as would be allowed as an equivalent.

2.1.10 Cargo Pump Room and Ballast Pump Room Ventilation

As mentioned earlier a brief mention should be made of the requirements for ventilation systems (as distinct from venting systems) and in the context of this paper these may be found in Part 5, Chapter 15, Sub-Section 1.7 and 1.8 of the steel ship Rules. Since these sub-sections are detailed to a greater extent than even the chemical tanker and gas carrier codes it is not proposed to examine them here but only to draw attention to the distinction which is now made between cargo pump rooms and other pump rooms. For example, a ballast pump room by itself requires only 20 air changes per hour. Such ballast pump rooms are found on some product oil tankers and many chemical tankers where a cargo pump room is unnecessary because each cargo tank has its own deepwell or submerged pump. However, if a ballast pump is fitted in the cargo pump room of say a chemical tanker carrying types A or B cargoes the number of air changes would increase to 30 air changes per hour because of the cargo pumps. Where, as indicated in 5, 15, 1.7.4, it is necessary for certain chemical cargoes to have 45 air changes per hour it is also necessary to have some sympathy for the man attending the cargo pumps who, in a cold climate, can freeze to death in the gale force winds or have his shoes and socks sucked off.

Conclusions on Part II

As far as inert gas systems are concerned the story is nearly ended and it is only as regards flame arresting devices that some loose ends need to be tied together in due course. A great deal of the interest in dealing with the two subjects described has been in attending the working parties at IMO, to see how IMO works and how highly efficient it can be in dealing with, and publishing, the reports on so many topics at each session. The very human and nationalistic elements are there too and the wonder of it all is that so many and varied persons can agree, or compromise, on anything at all let alone the mass of literature which eventually materialises. In this respect Resolution A500 at IMO has now restricted this outflow by stating that no further

subjects will be debated unless a definite need is shown to exist. So perhaps we can now look forward to having a period of stabilisation during which we can catch up with ourselves and gain experience in the new codes and practices before the next surge of knowledge bursts upon us.

ACKNOWLEDGEMENT

The author wishes to acknowledge the assistance of the Technical Illustrators for the preparation of the diagrams and also the following companies who supplied some of the illustrations used:

PRES-VAC ENGINEERING LTD. A/S
WILSON WALTON INTERNATIONAL (UK) LTD.

BIBLIOGRAPHY

1. Capt. W. D. J. Barker and T. W. Allsop: "Crude Oil Washing: Implementation and Operating Procedures". I.Mar.E., 14.10.80.
2. C. F. Day, Cdr E. H. W. Platt, I. E. Telfer and R. P. Tetreau: "The Development and Operation of an Inert Gas System for Oil Tankers". I.Mar.E., 1972.
3. "Inert Gas Systems". 1983 Edition IMO
4. "Regulations for Inert Gas Systems on Chemical Tankers". Resolution A567. (14) IMO.
5. "Standards for The Design, Testing and Locating of Devices to Prevent the Passage of Flame into Cargo Tanks in Oil Tankers". MSC/CIRC.373. IMO.
6. J. H. Burgoyne: "Accidental Ignitions and Explosions of Gases in Ships".

APPENDIX A

INERT GAS SYSTEM;

Scope of Survey of Various Items

It is considered that the scope of the Survey of the various items as listed should be as follows. The number opposite each item indicates the Survey required:

1. To be made under survey.
2. To be made under survey if 100 kW and above, works test certificate if less than 100 kW.
3. Works test certificate, and environmental test if not already carried out.
4. Environmental test if not already carried out.
5. Works hydraulic test certificate.
6. No survey required.
7. Survey after installation only, unless specially requested at works by customers.

The above refers to inspection at the Manufacturer's works only. It will, however, be necessary for the plant to be installed on board the ship and tested under working conditions to the satisfaction of the Surveyors in accordance with Part 5, Chapter 15, paragraph 7.8.1 of the Rules.

Furthermore, Annual and Special Surveys require to be carried out as follows:

Annual Surveys: See Part 1, Chapter 3, Paragraph 2.2.21.
Special Surveys: See Part 1, Chapter 3, Section 17.

If the blowers are driven by steam turbines or gas turbines (either as a separate gas turbine unit or as part of a gas turbine generating set) and the power exceeds 110 kW (150 shp) the turbines should be made under survey in accordance with the relevant sections of Part 5, Chapter 3 or 4.

For powers under 110 kW (150 shp) the turbine casing pressure test as per Part 5 Chapter 3 (or 4) Paragraph 6.3.2 (or 6.2.1) and a full load test of the complete unit, including overspeed and safety cut-out tests, should be witnessed. In addition, Manufacturers test certificates for the materials of the turbine rotor blading should be provided.

Table A1 List of Items for Survey

Item	Survey No.
Scrubber	7
Deck Seal	7
Heating Coils	5
Starters	6
Blowers (complete)	2
Scrubber water supply pump — motor	2
Scrubber water supply pump — pump	6
Valves, shipside	1
Valves, other than shipside	6
Main gas control valves	6
Blower isolating valves	6
Deck isolating valves	6
Recirculating valves	6
Boiler uptake valves with fail safe gear etc.	6
Main panel assemblies	1
Expansion bellows	5
Non return valves	6
Gas temp probe O_2 Indicator O_2 Analyser Pressure Switch	4
Press Transmitter Press Controller	3

APPENDIX B

INERT GAS SYSTEM;

Suggested Programme for Testing

1. Owing to their diversity, the Society has no specific test programme for I.G. systems and accepts the manufacturer's test procedure provided the requirements of paragraph 5,15, 7.8.1 of the Rules are complied with to the Surveyor's satisfaction.

2. However, for guidance purposes only, the following comments are made.

2.1. When examining I.G. system plans in Headquarters it is assumed that the manufacturer's stated capacity of the I.G. system and combined capacity of the blowers are in accordance with paragraph 5,15,7.2.4 and 5,15,7.4.1 respectively of the Rules.

2.2 It is concluded that the contents and/or requirements of each paragraph from paragraph 5.15.7.2.1 to 5.15.7.8.1 will have been considered and settled to the Surveyors' satisfaction before testing is commenced.

2.3 It is suggested that in co-ordination with the manufacturer's practice the following procedure may be followed for testing the I.G. plant:

On Deck

2.3.1 Check that the deck seal overboard discharge valve is open.

2.3.2 Check "low water level in deck seal" audible and visual alarm as per 5,15,7.7.7(g).

2.3.3 Check that there are no personnel in the tanks and advise the deck department of intended use of the plant on the selected tank.

2.3.4 Check that the pressure/vacuum breaking device (5,15,7.6.3) is filled with liquid to the correct level. This check should be made with the cargo tank at atmospheric pressure otherwise a false level will be created.

2.3.5 Check that the inert gas isolating valve or blank (5,15,7.5.15) on the selected tank is open and that the tank lid is securely closed and secured. The isolating valves on the remaining cargo tanks should be closed during the test.

2.3.6 Check that the lids if fitted on high velocity vents are closed and the pressure vacuum valves are correctly adjusted (5,15,4.2.5) In this connection it should be noted that the pressure setting of the P/V breaker should be higher than the setting of the P/V valve(s) and/or high velocity vent, if fitted.

In Engine Room

2.3.7 Set the boiler plant in operation and adjust the automatic combustion control (5,15,7.2.1) to maintain good combustion conditions producing a satisfactory Oxygen (0_2) content.

2.3.8 Check that the scrubber overboard discharge valve is open.

2.3.9 Check that the water supply to the scrubber and to the deck seal is in order.

2.3.10 Open shut-off valve(s) in outlet lines from the boiler uptakes to the gas scrubber.

2.3.11 Open blower(s) suction and discharge valves.

2.3.12 Start blower(s).

2.3.13 Keep automatically controlled valve (5,15,7.5.3) closed. Shut off water supply to scrubber and check that the blowers stop and audible and manual alarms, indicating low water pressure and blower failure, operate as per 5,15,7.7.7(a) and 7.7.9. Note that it should be impossible to start the blower(s) with the scrubber water supply shut off.

2.3.14 Re-establish water supply and restart blower(s) check that their alarms and lights go off panel.

2.3.15 Check that the 0_2 analyser and recorder (5,15,7.7.2) are in operation and that the 0_2 content is below the alarm level stated in 5,15,7.7.7(e)

2.3.16 Check that the 0_2 content of the inert gas line after the blowers is almost the same as the 0_2 content in the flue gas from the boilers since any appreciable increase after the blowers will indicate that air is being drawn into that part of the system.

2.3.17 Check "high oxygen content of the inert gas system" audible and visual alarms as per 5,15,7.7.7(e)

2.3.18 Check "high temperature of inert gas" audible and visual alarms as per 5,15,7.7.7.(c).

2.3.19 Check "failure of automatic control system power supply" audible and visual alarms as per 5,15,7.7.7.(f)

2.3.20 Open automatically controlled valve (5,15,7.5.3) and allow inert gas into deck main and selected tank.

2.3.21 Shut off water supply to scrubber and check that a) the blower(s) stop and b) the automatically controlled valve (5,15,7.5.3) closes and the inert gas recirculating or vent line valve opens. The audible and visual alarms for low water pressure and blower failure should operate together with those for low pressure of inert gas as per 5,15,7.7.7(h) and 5,15,7.7.14.

2.3.22 Re-establish inert gas supply to selected tank.

2.3.23 Checks should be made with a portable 0_2 analyser of the 0_2 content of the tank being inerted. In this respect it is suggested that sampling should be taken at three depths at different positions in the length and breadth of the tank, as considered necessary, to show that an inerted atmosphere exists at all parts of the tank. When the tank has been completely inerted it can then be pressurised to test any automatic pressure cut-out arrangements and the P/V valve(s) and/or high velocity vent checked for leakage and adjustment if necessary.

3. In the case of new ships paragraphs 4.6 and 6.5 of Part D1 of the Survey Procedures Manual would require a maximum capacity test under operating conditions. Obviously this is not easily arranged when carrying cargo so alternative suggestions are as follows:

3.1 With one or more cargo tanks empty and cleaned to a high standard fill with sea water and discharge to sea timing the discharge to equate with the capacity of the tanks and design capacity of the cargo pumps and at the

same time confirm that the I.G. plant is capable of maintaining an overpressure of inert gas in the cargo tanks.

Should such a test be carried out on an existing tanker, pollution aspects would have to be taken into account.

or

3.2 Where arrangements permit, fill and maintain inert gas pressure in cargo tank(s) such that the I.G. is discharged through, for example, a high velocity valve and then compare the pressure in the tank with the Manufacturer's capacity curves for the high velocity vent at the set pressure.

3.3 If neither of the above tests is satisfactory it would be expected that the IG Manufacturers would be able to suggest alternative ways of proving his equipment.

4. In the case of exisiting ships, paragraphs 3.13 of Part D2 of the Survey Procedures Manual requires only a working test, not a capacity test, to show that the sytem is satisfactory in operation.

Should, in certain circumstances, the cargo pump discharge capacity be in excess of the inert gas supply causing the pressure to fall there are always the two low pressure alarms (100 mm. WG as per 5,15,7.7.7.(h) and 50 mm. WG as per 5,15,7.7.14) to warn of this condition and allow the operator to reduce the speed of cargo discharge accordingly.

If an owner wishes a table could be affixed in the cargo control room to indicate cargo pump discharge capacities against pressure heads or to have a mechanical link up so that the maximum rate of discharge is restricted to 80% of the inert gas system capacity to ensure an over capacity of inert gas of 1.25 times the cargo pump discharge rate.

LLOYD'S REGISTER TECHNICAL ASSOCIATION

MINUTES OF THE 1986 ANNUAL GENERAL MEETING

The Annual General Meeting of the Technical Association was held in the Committee Luncheon Room on Wednesday, 28th May, at 1500 hours. Twenty three members attended.

The President of the Association, Mr. D. Rennie, occupied the chair:

AGENDA

1. Apologies for absence.
2. Approve and sign minutes of the last AGM.
3. Matters arising from the last AGM.
4. President's Review.
5. Hon. Secretary's Report.
6. Hon. Secretary for Corresponding Members' Report.
7. Hon. Treasurer's Report.
8. Proposed Syllabus for 1986/87 Session.
9. Election of Committee for the 1986/87 Session.
10. Newly elected Committee to assume office.
11. Any other business.

Each item on the Agenda was dealt with as follows:

ITEM 1

Apologies for absence had been received from Messrs. Finney, Carlton and Van Dorp.

ITEM 2

The Minutes of the last AGM, held on the 22nd May, 1985, were examined by the members present. The minutes were unanimously approved without amendment and signed by the Chairman.

ITEM 3

The Chairman asked the Hon. Secretary if there were any matters arising from the minutes.

The Hon. Secretary gave the following response concerning item 11;

The Committee has discussed the implications of Mr Wright's comments and considers that these are outside the scope of the Associations permissible activities since LRTA papers are published for private circulation among the Society's Staff only, in accordance with Rule 7.

For those cases where a wider circulation of a paper is thought appropriate, standing instructions are given in the Society's Circular No. 2356 about the procedures to be followed.

With respect to publication in 100A1, the authors of all papers retain the right of subsequent publication, subject to the sanction of the Committee of Lloyd's Register of Shipping, and this matter would thus be a subject only for the consideration of Publications Department and the authors concerned without reference to LRTA.

However, as is the case with all aspects of the Associations activities, this matter will be kept under review.

ITEM 4

In his review of the past year the President, Mr. D. Rennie, said:

"In the intervening period since the last AGM the Association has continued its efforts to promote the advancement and dissemination of knowledge among its membership.

A full programme of six technical papers and one guest lecture was successfully completed thus sustaining the level of activity of the previous year. The last two papers of the session were late changes to the published syllabus and I would express my thanks to the authors concerned for their assistance in providing suitable manuscripts at short notice.

The overall standard of papers being presented remains high and the Society's expertise in both general and specialist areas has been well demonstrated. This fact has also been recognised by Lloyd's Register by the selection of three LRTA papers for Technical Reprints during my two years of office.

It has been pleasing to note that attendances at meetings during the last year have improved. However, the Association will always be happy to accommodate even more members, particularly at the presentation of some specialist papers—none of us are to old to learn something new!

In the latter part of the session revised arrangements were made with respect to the discussion period following a paper's presentation. It was intended that the limitation of a maximum five minutes for each contributor would both encourage and achieve a more active involvement from the floor. I feel that this was most notably successful at the last paper in the session given by Clive Bainbridge.

This marks the end of my two years as President and also the end of the 66th year of the Association. At the foundation in 1920 the first President said that "the scope for papers was large and the subjects numberless". Having regard to the diversity of work upon which Surveyors are now called to engage, there is probably as much scope today, if not greater than at any time in the past, for papers on an even wider variety of subjects.

Finally, I would like to thank the Officers and Committee Members for all the hard work put into the organisation of the Association's affairs and trust that you will have confidence in Les Beckwith to see that the Association continues to thrive in the forthcoming year."

ITEM 5

The President called upon the Hon. Secretary to report on the Association's activities during the Session. Mr. Magill advised the meeeting as follows:

The Technical Association has had a very active year in which six technical papers, and a guest lecture were presented. The attendance at meetings throughout the year was generally higher than has been experienced in previous sessions and the Committee Members are thanked for their efforts in promoting LRTA to achieve this increase.

PAPER 1. At the first meeting in October, 1985, Mr A. Lough presented his paper "Dynamic Positioning" to an audience of 56 members.

PAPER 2. At the November, 1985, meeting Mr R.V. Pomeroy read the joint author paper by Messrs. Cameron and Pomeroy entitled "Fracture Mechanics: An Introduction and Review" to an audience of 93 members from a wide spectrum of the Society's specialist disciplines.

PAPER 3. In December, 1985, Mr A. McInnes from the Yacht and Small Craft Group at Southampton presented his paper "Some Hull Construction Aspects of Small Patrol Boats" to an audience of 44 members.

Guest Lecture This lecture was given in January 1986 by Prof. Sir Hugh Ford, the subject being "Welding and Design Aspects of Offshore Structures". The meeting was attended by 112 members and the skilled delivery by Sir Hugh was followed by a lively discussion period. The draft text of the lecture was not submitted by the author, therefore there is no published record for this meeting.

PAPER 4. At the February, 1986, meeting Messrs. Kunz and Syed presented their paper "Calculation of Crankshaft Stress and Service Measurements". The meeting was attended by 56 members.

PAPER 5. The March, 1986, meeting saw a change to that published in the syllabus. Mr. A. Sokolov, Lowestoft Office, presented a paper entitled "Diesel Engines in Hazardous Areas on Offshore Installations" to an audience of 35 members.

PAPER 6. The final meeting of the session in April 1986 was also a change to that indicated in the syllabus. Mr C. A. Bainbridge presented his paper "Strength and Fatigue Analysis: the North Sea Experience" to an audience of 92 members.

The Association is very grateful to the authors of these last two papers for the way in which they were able to provide suitable draft manuscripts for publication at very short notice.

The LRTA Medal for the 1984/85 Session was presented to Messrs. C. M. Magill and D. J. Holland for their paper "Fire Safety Aspects of the 1981 Amendments to SOLAS '74".

As previously mentioned, the attendance at meetings has shown a very encouraging upturn, being on average 35% above that of the previous session. Pleasing as this may be it still only represents about 13% of the membership resident in HQ and nearby Offices (9% last session) and it is hoped that there is still room for a further improvement in the coming session.

Cursory perusal of the attendances at meetings has shown a higher proportion of Engineer Surveyors attending than Ship Surveyors which reflects the general trend of more "engine disciplines" papers than "ship disciplines". If the "ship" members are not attending because of the lack of "ship" papers, then please may we have more offers of papers from the ship disciplines—those presently being received are few and far between!

ITEM 6

The President then called upon Mr. Magill to give the report on behalf of the Assistant Secretary for Corresponding Members. The meeting was advised that:

The Asst. Secretary for Corresponding Members retired from the Society earlier this year and I have been looking after these duties in the interim period.

Regretably there is little Outport activity to report on. A circular letter addressed to all the UK Regional Managers encouraging them to hold an LRTA Meeting in their area, similar in format to the very successful

meeting held in Newcastle last year, has received only one response. It is proposed to hold a meeting in Southampton in late summer, about the time of the Southampton Boat Show, to which selected clients will be invited. However, this will take place during the next session and will be reported at the next AGM.

The Corresponding Members maintain good contact with the Association and there is little difficulty in obtaining the LRTA Medal adjudication forms for example. Thus it appears that sufficient interest in our activities does exist outside the London area and it is incumbent upon the Committee and the Corresponding Members to instigate means whereby the dissemination of knowledge among members is improved during the coming year.

There are two changes to record this year among the Corresponding Members:

1. Mr. K. J. Frier at London and District on his transfer to Croydon.
Nominee: Mr. C. D. Wilkie.

2. Mr. A. R. Morton at Glasgow on his retirement from the Society.
Nominee: Mr. J. F. Cooper.

ITEM 7

The President then asked the Hon. Treasurer to present the statement of the Association's finances during the past year. Mr. Lindsay presented the financial statements reproduced at the end of the AGM Minutes.

No comment was made by the members present.

ITEM 8

The President asked Mr. Beckwith the Association's Vice-President and Chairman of the Sub-Committee on Technical Papers to present the proposed syllabus for the 1985–86 Session. Mr. Beckwith advised the meeting as follows:

"The Sub-Committee on Technical Papers met four times during the 1985/86 Session for the purpose of developing balanced programmes for the forthcoming sessions.

A full programme of six technical papers and a guest lecture is proposed for the 1986/87 Session as follows:

Paper No. 1 Inert Gas and Venting Systems
by B. W. Oxford of M.D.A.

Paper No. 2 The Resurrection of Passenger Ships
by J. J. Stansfield of Piraeus

Paper No. 3 Heated Submarine Pipelines
by Dr. D. Richards of OED.

Paper No. 4 Sterntube Bearings, Performance Characteristics and Influence on Shafting Behaviour
by R. W. Jakeman of A.E.S.

Paper No. 5 Strength Analysis of Self-Elevating Platforms
by W. J. Winkworth of OSG.

Paper No. 6 Materials and Welding for Offshore Structures
by C. S. Whitcroft also of OSG.

The Guest Lecture on January 14th 1987, is to be given by Professor Stanley of Manchester University whose specialist field is stress analysis of pressure vessels.

It is additionally proposed to publish a discussion paper on Hull Attachments by Mr. Komori of Hiroshima, who has recently retired from the Society.

Looking further into the future the subjects under consideration for the 1987/88 Session are:

The Society's involvement with International Conventions,

Vessel motions and their effect on lifting operations at sea,

Propellers,

Hull structures aspects of container ships,

Electric propulsion, including the QE2 re-engining.

Electronic Mail and communications in the Society.

The Sub-Committee moved the adoption of the proposed syllabus for the 1986–87 Session.

There were no comments from the members present and the proposed syllabus was duly ratified.

ITEM 9

The President requested the Hon. Secretary to advise the AGM on the election of Officers to the Committee.

Mr. Magill read out the list of names nominated for the Officers, Members of Committee and Corresponding Members of Committee. In each case there was only one nomination for each office and the nominees were duly elected. The names of those elected are recorded below:

President	:	Mr. L. Beckwith.
Vice-President	:	Mr. D. McKinlay.
Hon. Secretary	:	Mr. C. M. Magill.
Hon. Asst. Secretary	:	Mr. A. W. Finney.
Hon. Asst. Secretary for Corresponding Members	:	Mr. J. Frize.
Hon. Treasurer	:	Mr. B. P. Sharman.

COMMITTEE MEMBERS:

Classification Reports	Mr. D. B. Parkin, Mr. D. Martyn
Hull Structures	Mr. T. Lindsay, Mr. P. Stringer
Industrial Services	Mr. E. D. Stephenson
International Conventions	Mr. F. R. Hales
MDAPAD	Mr. R. Jeffrey
Offshore Services	Mr. D. Harris
Refrigeration	Mr. J. Rushmer
Research Laboratory	Dr. F. A. Khayyat
TID/AES	Mr. N. A. Hall-Stride

CORRESPONDING MEMBERS OF THE COMMITTEE:

AUSTRALASIA	Mr. R. B. Last	SYDNEY
BELGIUM AND FRANCE	Ir. W. J. G. de Backer	ANTWERP
CANADA	Mr. C. A. C. MacGregor	MONTREAL
CENTRAL AMERICA AND THE CARIBBEAN	Mr. F. Klug	MEXICO CITY
CENTRAL EUROPE	Mr. C. M. Bergmann	HAMBURG
	Mr. D. Terno	DUSSELDORF
CENTRAL ORIENT	Mr. J. N. Mckay	BUSAN
EASTERN MEDITERRANEAN	Mr. J. J. Stansfield	PIRÆUS
IBERIAN PENINSULA	Mr. H. Garcia	MADRID
INDIA AND PAKISTAN	Mr. S. V. Ramchandani	BOMBAY
ITALY AND MALTA	Dott. Ing. G. Perrotta	GENOA
JAPAN	Mr. A. Kamitami	OSAKA
	Mr. R. Hashiguchi	SHIMONOSEKI
	Mr. K. Seki	YOKOHAMA
MIDDLE EAST AREA	Mr. S. M. Miskry	BAHRAIN
NETHERLANDS	Ing. A. Schreuder	ROTTERDAM
NORDIC COUNTRIES	Mr. J. G. Lassen	COPENHAGEN
	Mr. R. S. Malmberg	GOTHENBURG
NORTH AFRICA	Mr. S. M. A. Ahmed	ALEXANDRIA
POLAND	Mr. A. Barrett	GDANSK
SOUTH AMERICA	Mr. R. De. F. Gomes	RIO DE JANEIRO
SOUTHERN AFRICA	Mr. C. A. Timms	DURBAN
SOUTHERN ORIENT	Mr. B. H. Wong	SINGAPORE
UNITED KINGDOM:		
EAST MIDLANDS	Mr. H. Milne	HULL
NORTH ENGLAND	Mr. A. J. O'Connell	NEWCASTLE
NORTH SCOTLAND	Mr. W. F. Rogerson	ABERDEEN
SOUTH EAST	Mr. C. D. Wilkie	LONDON
SOUTH SCOTLAND	Mr. J. F. Cooper	GLASGOW
SOUTH WEST AND SOUTH WALES	Mr. D. G. Taylor	SOUTHAMPTON
WEST MIDLANDS AND EIRE	Mr. D. G. Gaskell	LIVERPOOL
UNITED STATES OF AMERICA	Mr. W. E. Tuck	NEW YORK

The Hon. Secretary also advised the meeting that the Honorary Auditors, Messrs. Leighton and Mounch, were willing to act in this capacity for the 1986–87 Session. The meeting then re-elected Messrs. Leighton and Mounch to continue in office for a further year.

Item 10

At this point in the meeting the retiring officers stepped down and the new members of committee took up their office.

Mr. Beckwith, the L.R.T.A's President, introduced the Session by expressing the L.R.T.A.'s thanks to those Committee Members who had retired and gave a welcome to those who had taken up office.

At this point in the preceedings the retiring President, Mr. Rennie, was presented with a humourously worded certificate prepared by the Committee in which recognition was given to the speed and firm control with which business had been dealt with at Committee meetings during his term of office.

Item 11

There being no further business the President closed the meeting at 15.40 hours.

L. Beckwith	C. M. Magill
President	Hon. Secretary.

LLOYD'S REGISTER TECHNICAL ASSOCIATION

Income and Expenditure Account for year ended 31 March, 1986

	£	£
Sale of 206 binders		445.40
Less:		
Cost of binders Sold	451.14	
Less:		
Binders subsequently found etc. after being written off in 1985 (Note 1)	61.32	
		389.82
Surplus		55.58
Less:		
Expenditure		
Gratuities	90.00	
Entertainment	59.00	
Engraving	17.50	
	166.50	
Less:		
Bank Charges refunded	5.25	
		161.25
Excess of Expenditure over Income		(105.67)
Grant from Society		300.00
Net Surplus		**£194.33**

Note: 1	The net figure of £61.32 is made up as follows:			
	Binders found	40 @ 2.19 =		87.60
Less:	Damaged or Lost in Transit	6 @ 2.19 =	13.14	
	Binders unaccounted for in 1986	6 @ 2.19 =	13.14	26.28
		28		61.32

Cash Budget for 1986–87 is as follows:

	£
Balance of cash 1.4.86	446
Plus Grant	300
	746
Less: Cost of entertaining, gratuities plus margin for contingencies	250
Estimated surplus	**496**

Note: 1. It is assumed that cost of postage will continue to be borne by the Society.

2. The volume of binder sales is unpredictable, therefore, no account of this has been taken in the above figures.

3. No account has been taken of the repayment due to the Society in respect of the balance of loan remaining of £1,350.

D. Leighton, F.C.C.A.
M. Mounch, F.C.C.A. } Joint Auditors

6th May, 1986.

LLOYD'S REGISTER TECHNICAL ASSOCIATION

Balance sheet as at 31st March, 1986

1985 £			1986 £	1985 £		1986 £
541.46	General Fund Balance B/F	507.52		2,285.82	Cash at Bank (Current A/C)	431.58
				—	Cash in Hand	15.00
(33.94)	Plus Net Surplus/(Deficit)	194.33	701.85	1,995.09	Stock of ring binders in hand at cost	1,605.27
507.52						
1,600.00	Loan from Lloyd's Register		1,350.00			
2,173.39	Sundry Creditors		—			
4,280.91			2,051.85	4,280.91		2,051.85

N.B. Stock has been valued at lower of cost or net realisable value.

Report of the Auditors to the Members

The above Balance Sheet and attached Income and Expenditure Account has been drawn up from the documents and vouchers of the Association and the explanations given by the Officers thereof. We have examined the Income and Expenditure Account and the Balance Sheet and are satisfied that they show a true and fair view of the activities of the Association for the year ended 31st March, 1986.

D. Leighton, F.C.C.A.
M. Mounch, F.C.C.A.
} Joint Auditors

6th May, 1986.

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Lloyd's Register Technical Association

Discussion

on the paper

INERT GAS AND VENTING SYSTEM

by

B. W. Oxford

This Paper was awarded the 1986/87 L.R.T.A. Medal

Paper No. 1. Session 1986–87

Any opinions expressed and statements made in this Discussion Paper are those of the individuals.

Hon. Sec. R. V. Pomeroy
71 Fenchurch Street, London, EC3M 4BS

Discussion on the Paper

INERT GAS AND VENTING SYSTEM

by

B. W. Oxford

Author's Note:

Subsequent to the galley proofs of this paper being edited the rules were changed by Notice No. 1, 1986, which, amongst other things, deleted any reference to chemical tankers in Part 5, Chapter 15 and placed them in a separate rule book. It should be noted therefore that references to chemical tankers in this paper still refer to the chapter, section and paragraph numbering as existed in the rule book before amendment.

Corrigenda:

The following errors have been identified and are offered for amendment.

1. Page 4 Paragraph 1.1.1 12th line. Amend in brackets to read "(See also 1.1.8)".
2. Page 4 Paragraph 1.1.2 10th line Amend "Section 17" to "Section 18".
3. Page 7 Paragraph 1.1.8 3rd line. Amend "1.1.2" to "1.2.1".
4. Page 7 RH column 2nd line. Amend "4,9,7.1.3" to "4,9,7.1.4".
5. Page 7 Figure 5 LH side. Amend "(25000m^3/w)" to "(25000m^3/hr)".
6. Page 7 Figure 5 RH side. Amend "5,15,7.6.1" to "5,15,7.5.3".
7. Page 9 RH column, 25th line from bottom. Amend "deadweight" to "deadweight/hr".
8. Page 9 RH column, 1st and 2nd lines from bottom. Delete "5,15,7.7.6(g)" in brackets.
9. Page 12 Figure 10, lower table.
   Amend = "3.5kg" to "3.5kg/10000m^3 of gas"
   = "0.08 kg" to "0.08 kg/10000m^3 of gas"
10. Page 13 LH column, 23rd line from top. In brackets insert full stop between 7 and 3 making it 62.7.3.
11. Page 13 RH column, 13th line from top. Delete "comma" after "dioxide" and inserts "and" and delete "or nitrogen"
12. Page 16 LH column, bottom line. In brackets amend 62.11.1 to 62.11.2.1.
13. Page 17 Figure 13. At top in brackets delete "(See Rules 5.15.7.15).
14. Page 17 Figure 13. Insert suffix '2' under 'P' to give "P_2 = ?" and
   $$\text{“}P_2 = \frac{1 \times 10000\text{”}}{6000}$$
   insert "absolute" between "the" and "pressure".
15. Page 20 RH column, 16th line from top. In brackets amend "62.19.12" to 62.19.1.2".
16. Page 20 RH column, 26th line from top. Amend 5,15,7.7.4 to 5,15.7.4.
17. Page 21 RH column, 27th line from bottom. Amend the word "Introduction" to "History".
18. Page 21 RH column, 12th line from bottom. Amend "Section 17" to "Section 18".
19. Page 22 RH column, 11th line from top. Amend "L.W." to "L.W.L.".
20. Page 24 LH column. In 8th and 14th line from bottom. Change "A473 (12)" to "A473 (XII)".
21. Page 33 LH column, 6th line from bottom. Change "Section 17" to "Section 18".

In MSC/CIRC. 373 attached to the paper. In paragraph 1.2.2 change "3.3.3" to "3.3.3.2".

DISCUSSION

In anticipation of the following verbal and written discussions the Author wishes to thank all those who were able to attend the lecture and in particular those who made contributions from both home and abroad.

From Mr. G. Coggon:

First of all I would liked to congratulate Mr. Oxford on an extremely good paper on two subjects of considerable importance to the Society. My only criticism is that it would have been even better if surveyors had been given this detailed information on such complex matters a little earlier.

The paper draws attention to the role of IMO in developing the regulations on inert gas and cargo tank venting, but I feel that recognition should also be given to the work done by the International Chamber of Shipping (ICS) on both subjects, without which IMO would have had great difficulties in starting their development, let alone completing them.

Representatives at Lloyd's Register do a lot of valuable work at IMO and this has been acknowledged there from time to time. However, in the realm of inert gas and tank venting we learned infinitely more than we gave. On the subject of inert gas the expertise came from the tanker industry through ICS and in the case of venting systems it was the Federal Republic of Germany delegation which seemed to have the preponderance of knowledge. I feel Mr. Oxford would agree that he himself learnt much by attendance at IMO.

We are most grateful to him for sharing his knowledge with us in this clear and interesting paper.

Knowing that I was committed to leading the discussion I have searched the paper for technical errors of omissions to comment on, but I have been unable to find any. However, I would like to comment on one or two items.

Firstly in the second paragraph on page 3, Mr. Oxford suggests that the 100,000 tons and 50,000 tons parameters mentioned in the previous paragraph were chosen on the basis of tank washing machine capacity. This was not so; they were chosen at a time of great urgency to improve the safety of VLCC's in the light of the 3 explosions which had taken place in the rapid succession in the MACTRA, MARPESSA and KONG HARRKOV VII and others in some combination carriers. It was essential to get the regulations applied to them and then consider extending them to other tonnages later and that is exactly what happened.

In paragraph 1.2.3 under the heading Classification Rules, Mr. Oxford slips in the briefest reference to our notation IGS. Now he may wish to soft pedal it, but I feel it should be highlighted. I consider that although it was useful at the time of its conception to indicate that a ship had an LR approved inert gas system, it is now deceiving to do so in many cases. This is because it is not the practice of LR to apply rule amendments retrospectively. Consequently there are many older ships with inadequate IG systems which have the IGS notation. Such systems could even be dangerous in that they lead an operator to believe that he has a safer ship than is the case or that it complies with SOLAS requirements. As stated in the paper SOLAS requires the same for an existing ship as for a new ship with only a few exceptions.

From Mr. J. R. G. Smith:

May I congratulate the Author on an extremely comprehensive and well written paper. It will surely become a standard reference on the subject.

I wish to confine my further remarks to Section 1.1.13 regarding inert gas on a liquefied gas carrier. The Author describes the by-pass of the deck water seal when in the liquefied gas carrying mode and the connection necessary when the ship changes to the naphtha carrying mode. Could the Author confirm that it is the responsibility of the crew to disconnect and reconnect as necessary when changing cargo modes, and that verification by a surveyor is not required?

Could he also confirm however, that during normal surveys of the system the surveyors satisfy themselves that the facilities to permit such change over procedures are readily available, having previously established from the Register Book that the ship is permitted to carry naphtha as well as liquefied gas?

Could the Author advise whether change over facilities are specially reported?

From Mr. J. S. C. Bloomfield:

Mr. Oxford's paper will be welcomed by the working surveyors. Here is a complete reference on IG and Venting in one excellent package.

In the Introduction Mr. Oxford states "the Society's Rules are written for NEW ships". This must be remembered at all times in T.O.C. cases and particularly when viewing that beloved 20 year old with one of the Owner's oldest representatives.

Figure 16 draws my criticism. The remote controlled sea valve is correct but the extra spool pieces and valves on the inboard side are to me, unnecessary. The knowledge and manufacture of GRP piping has improved tremendously since the early days and Mr. Oxford may wish to comment on this.

WRITTEN DISCUSSION

From Mr. R. Moore:

It is a fundamental principle of safe tanker operation that when it is required to gas free a tank after washing it should first be purged with inert gas to reduce the hydrocarbon content to not more than 2% by volume so that during the subsequent gas freeing operation no part of the tank atmosphere is brought within the flammable range. This is clearly demonstrated by the flammability diagram—Fig. 3 in the paper.

The Rules require portable instruments for measuring oxygen and flammable vapour concentration to be provided but do not make clear that the hydrocarbon content must be measured with an appropriate meter designed to measure the percentage of hydrocarbon in an oxygen deficient atmosphere. The usual flammable gas indicator is not suitable for this purpose. For guidance, the Surveyors who may be expected to verify that portable meters are provided, the types available are adequately described in the book "International Safety Guide for Oil Tankers and Terminals". viz:

Catalytic Filament Combustible Gas Indicator—used for hydrocarbon gas in AIR at concentrations below the lower flammable limit (LFL) with scale marked either with

% LFL (lower flammable limit) or

% LEL (lower explosive limit).

Such indicators must not be used for measuring hydrocarbon gas in inert atmospheres.

Two types which can be used in inert atmospheres and which are available commercially are:

(a) Non-catalytic heated filament gas indicator
(b) Refractive index meter.

The scale in these cases is marked in percentage by volume of hydrocarbon gas.

In view of the foregoing is it not time that guidance for the surveyors should be issued or should not the Rules be amended to make it clear that a hydrocarbon meter available for use in an oxygen deficient atmosphere is required?

From Mr. P. Stanney:

I would first of all like to congratulate Mr. Oxford on his most comprehensive and informative paper. As an ex-Inert Gas commissioning engineer now employed by the Society I would like to offer the following comment:

My experience with Inert Gas Systems mainly concerns the conversion of existing medium sized crude carriers. These vessels constitute a large sector of the total tonnage and whilst Inert Gas Systems fitted on newbuildings appear to give no particular problems those fitted retrospectively seldom perform as well as the designer intends. This is for a variety of reasons including the limitations imposed on the position of the major equipment and the more strict financial constraints but mainly because of the following.

The widespread impression is that in order to produce Inert Gas it is only necessary to connect up to a boiler uptake and out will come suitable flue gas. In practice this is far from the case. In the days of steam VLCCs it was normal to produce 5% or less Oxygen in the uptake with the boiler on 75% or more load. However, the more modern diesel tankers derive their Inert Gas from auxiliary boilers with much smaller furnaces and less sophisticated combustion control equipment when it may be difficult to obtain low oxygen flue gas on anything less than 100% load and then only by reducing the air for combustion to levels where large quantities of soot are formed. During system start-up prior to discharging cargo the boiler may be on a very light load and large quantities of soot will inevitably be carried into the scrubber.

Newbuilding tankers can of course be fitted either with special fuel burning equipment or, if the furnace size is too small, a separate Inert Gas Generator burning diesel fuel (an expensive alternative) may be the answer. However, what of all the existing medium sized motor tankers which do not have suitable fuel burning equipment and may not have sufficient space to be fitted with a Generator?

The answer is unfortunately that whilst it may be possible to operate the system initially it will not be long before the production of excessive soot causes problems. The Scrubbing Tower will have a rating to remove a fixed percentage of both solids and S02/S03 gas and whilst the S02/S03 gas can easily be removed by simply cooling the uptake gas this is not true for any excessive solid content. A large proportion will be carried through the Scrubbing Tower and into the low temperature side of the system and deck pipework, thus effecting not only the system valves but also the instrumentation. On many occasions I have been forced to wonder whether a vessel would be safer without its IG system than with a system suffering from a jammed non-return valve, chocked flamescreens and corroded pipework.

Would it not be a relatively simple matter to monitor the uptake gases during the first system survey for 02 and the solids contents with the boiler on say 50% load to establish the boilers suitable for the production of Inert Gas.

From Mr. H. J. M. Jonkers:

Chemical tankers, in general, need dry water seals. Apart from the example shown in the paper we have seen on some newbuildings a block and bleed system where, apart from the most forward non-return valve, a combination set of double operating automatic isolating valves and a double set of bleed valves to atmosphere are fitted.

Since this system has its own attractions I wonder whether it is considered as being fully in compliance with, or regarded as equivalent to, the LR Rules and IMO regulations.

Regarding power failure referred to in paragraph 17.1.6 of

Resolution A.567(14) it is my opinion that, in addition to indication of power failure on the indicating devices as referred to in paragraph 14.1, power failure indication must also be provided on the oxygen monitoring equipment.

We had the experience in a newbuilding that the whole plant could operate with the oxygen analyser switched off since zero oxygen reading at that moment was just for the perfect indication to start the system. It sounds stupid but it happens. So any suggestions would be appreciated.

Another effect also created some problems. The scale of the instrument could be altered by a scale switch and it is considered therefore that the alarm function should be independent of any reading or change of scale switch. The oxygen repeater in the cargo control room also depends upon the scale of the master instrument. Again in the control room there was no automatic reference to the scale position.

With regard to chemical/grain carriers these are required to have the inert gas pressure monitored on the isolated slop tanks when on intermediate, ballast or grain voyages. Again a power failure alarm for this should be incorporated in the bridge equipment. In the case of UMS ships it can be linked with the engine room monitoring system.

If we now consider the maintenance of inert gas plants, at annual surveys on tankers or combined tankers the surveyor is supposed to get an idea about proper maintenance and clear the installation. In my opinion IMO and classification should insist that maintenance records are kept on board especially in respect of PV valves. A spare valve of each type on board could be used in sequence for maintenance and also for verifying the settings when installed on a slop tank test flange. Records of this would at least help the surveyor in his judgement.

In connection with the isolation of individual tanks from the inert gas main the use of spectacle flanges is allowed. Personally I prefer the double isolating valve since I have my doubts about spectacle flanges being used in practice. The only one which is foolproof is the one shown in Fig. 12 but these are matters of opinion and the responsibility has to be taken on board.

From Mr. R. M. Hobson:

First of all my congratulations to Mr. Oxford upon the thoroughness with which his paper covers the subject matter.

Upon first reading of the paper I was aghast at the many safety features he refers to and which were undoubtedly missing on many of the systems I have dealt with in the past.

However, I take heart from his opening remarks wherein he states that Inert Gas and Venting Systems have been passing through an evolutionary period over the last few years. Most of us choose less polite terminology for the proliferation of Rule requirements.

The paper gives me an opportunity to ask a question on an aspect of IG systems that has long puzzled me!

I refer to the use of spade blanks to isolate individual cargo tanks from the inert gas main. A typical example is shown on page 16 of the paper, Fig. 12.

Occasions arise in the operation of a tanker when it is necessary to isolate just one or two cargo tanks from the IG system—for example when tank inspection and/or mucking out of sludge requires to be carried out progressively tank by tank. This can only be effected, I believe, by inserting the spade blank whilst the tank is already full of inert gas.

When one visualises a cargo tank hatch, its coaming height, the position of the IG pipe and so on, I wonder if it is really practicable? The operation must be fraught with the possibility of asphixiating the crew member concerned and personally, in the case of ships coming under my own supervision, I have always been able to persuade the Owners to fit isolating valves in the deck line between the IG main and the cargo hatch.

Mr. Oxford touches in his paper upon the purpose of pressure-vacuum valves in his explanation on page 16 of paragraph 5.15, 7.5.17 of the Rules which requires cargo tanks to be protected against overpressure or vacuum caused by thermal variations, allowing for example, the tank to breathe out during a hot day and to breathe in during a cool night.

This aspect is referred to more fully on page 26 of the paper and an example of a typical pressure vacuum valve (combined with a high velocity vent) is shown in figure 21(a).

An alternative example is shown in figure 21(b) but in this case there is no provision for automatically releasing the small volumes of vapour/air mixture flowing during a normal voyage except through the high velocity section of the valve.

Does the valve (figure 21(b)) really cope effectively with its two alternative duties?

(a) High velocity venting of large quantities of gas during cargo loading and

(b) The slow build up of internal pressure in a tank due to climatic variations in ambient temperature.

I have heard it said that the sudden and unexpected "popping" of these valves during a tranquil period of the day can be most disconcerting to a watchkeeping officer with his mind on other matters. Indeed Mr. Oxford perhaps unwittingly refers to this aspect when he states on page 27 of his paper that the valves have to be tested to show that they will only open when an efflux of 30m/sec can be attained.

This brings me to another matter. The high velocity valve will be set to open at perhaps 1400 mm water gauge and when tank loading is commenced the pressure will build up in the cargo tank until the high velocity valve "pops" off its seat at this pressure. But due to the effect known as "accumulation of pressure" the vapour pressure in the tank may build up to a higher pressure to the point where the pressure breaker on the inert gas main has to relieve the excess pressure.

By way of example a typical high velocity vent may pass 1650 cu. metres of vapour at its opening pressure of 1400 mm water gauge whereas if loading rate is increased to 2000 cu. metres per hour then the high velocity valve will be relieving a pressure build up of nearly 2200 mm water gauge.

Possibly a bigger valve, or two valves per tank should have been chosen in such an example.

Does the Society in fact take any interest in loading rates relative to the venting arrangements fitted?

Lastly, one light hearted comment on page 9 of the paper disappointed me. Mr. Oxford refers to the possiblity of discharging a ship's oil cargo to storage tanks half way up Mount Everest!

Lloyd's Surveyors seem to have a fixation with Mount Everest and I recall that Mr. F. H. Atkinson presented a paper on the hull surveys of VLCC's in which it was claimed that the effort in surveying the tanks of such a ship was equivalent to climbing Mount Everest.

Surely if Surveyors can climb to such a peak then with the aid of our machinery we can pump oil to the top and not just half way up?

AUTHOR'S REPLY

To Mr. G. Coggon:

The Author agrees that he has learnt much by attendance at IMO but also wishes to give credit to Mr. Coggon for his introduction of the Author to the International Chamber of Shipping, IACS and finally to IMO and for his help and encouragement at the various meetings they have attended together.

With regard to Mr. Coggon's comment that it would have

been better had the surveyors been given this information a little earlier it is perhaps, as Mr. Coggon will know, that there has been much confusion about how and to what extent the requirements contained in Regulation 55 of Chapter II-2 of the 1981 Amendments regarding the inert gas and venting systems on board oil tankers should also apply to chemical tankers and gas carriers. As mentioned under item 2.1.8 of the paper the venting arrangements on chemical tankers have still not been finalised.

It is agreed that in the context of British Ships the International Chamber of Shipping, with all its combined expertise from the tanker industry, is the leading light in initiating discussions on these subjects for presentation at IMO via the Department of Transport representatives.

With regard to the parameters of 100,000 and 50,000 tons mentioned in connection with the requirements for inert gas systems to be fitted on tankers and combination carriers of these sizes after the explosions on the three VLCC's mentioned, the Author would bow to Mr. Coggon's superior knowledge and involvement at that time but would nevertheless stress the emphasis that was placed on the generation of static electricity from tank washing machines as was evident in the reports subsequent to those explosions.

Regarding the IGS classification the Author had no wish to soft pedal on the IGS notation and for many years he wondered why an Owner should have had the option of having an IGS notation or not. Nevertheless, from the time that inerting systems were first fitted on board tankers classed with the Society careful consideration has been given to such systems and to the safety aspects involved as is in evidence by Circular 2381 and, in fact, it could well be said that SOLAS's rules for inert gas from 1973 to the 1981 Amendments were less comprehensive than those of the Society and also were not retroactive for existing ships.

It is perhaps obvious and perhaps unavoidable that a time element often comes into play between the application of SOLAS regulations and the Society's Rules and again this shows the importance of Surveyors attending IMO and other bodies in order to keep up to date with current thinking and incorporating that knowledge in replies to any verbal or written discussions.

To Mr. J. R. G. Smith:

Being a near resident in British West Hartlepool at some time in the past it is always a pleasure to have a "Geordie" asking these awkward questions.

Although the Society originally turned down the idea in 1971 of carrying Naphtha, which is a non-boiling cargo, on liquefied gas carriers it was later allowed provided the Naptha was carried in inerted tanks, the inert gas being at a positive pressure at all times during carriage and also when discharging. In addition alarms were to be provided to cut out the cargo pumps at low inert gas pressure to avoid air being drawn into the tanks.

Since at that time this was a new venture the Society was naturally cautious and one of the requirements was that the proposed change-over arrangements should be checked, before and after the carriage of Naphtha, by the Society's surveyors and each voyage treated on an individual basis until satisfactory experience was obtained.

This experience having been obtained over a period of years with satisfactory results it is now the responsibility of the crew under the supervision of a responsible person on board to ensure that the change-over arrangements are in the correct mode. If the Society is requested to approve a liquefied gas ship for the carriage of other non-boiling cargoes or if the ship is of a novel design it would obviously be prudent for a surveyor to attend the first loading to check the arrangements on a prototype basis.

On investigation it would appear that there are no special provisions for reporting these change-over facilities and one of the reasons for writing this paper was to draw attention, for the surveyor's guidance, to items of which he may be unaware since these specific requirements are not included in the Rules.

Nevertheless, it should be emphasised that a letter detailing the arrangements to be complied with for the carriage of Naphtha is given in each initial approval and it is hoped that surveyors, having checked the ships notation, will now be aware of the special arrangements to look for.

To. Mr. J. S. C. Bloomfield:

In our consideration of ship safety from pumping and piping aspects in Head Office we are particularly concerned about fire and flooding and both aspects are present in Fig. 16.

As previously mentioned in the paper GRP has a low melting temperature and should the effluent overboard discharge be burnt out flooding would take place from both the scrubber (until the sea water pump is shut off) and the sea. Since the GRP was fitted in the first place to combat corrosion and erosion it may be that the metallic shipside valve has corroded and, by itself, could not stop flooding from the sea. So the non-return valve is fitted as the first reserve to preserve the integrity of the shell plating and the intermediate drains are provided to indicate the continued efficiency of the ship side valve, or otherwise, so that should it prove defective by the presence of water at the drains the ship can either be docked at an earlier date than might have been arranged for or the crew, at least, will be aware of the defect and possibly be able to make some precautionary arrangements.

The Author cannot think of any other discharge where the effects of corrosion and erosion are so evident and where precautions additional to the normal have to be adopted.

To Mr. R. Moore:

Mr. Moore's comments are very valid.

In reply it would be expected that the suppliers of the inert gas system would provide the appropriate meters in the initial package but, being portable, such instruments are liable to be damaged or lost. However, the Society cannot cater for operational features other than to ensure that at the appointed survey periods these devices are still available and in good working order. In other words it is the Owner's responsibility to obtain suitable replacements but it is important that the surveyors should be aware that some flammable gas indicators are not suitable for use in an oxygen deficient atmosphere not forgetting that not all tankers, for example those below 20,000 tons deadweight, require to be fitted with inert gas.

It is hoped that both the paper and this discussion paper will draw surveyors' attention to these points or that mention may eventually be made in the Survey Procedures Manual.

Also Mr. Moore has mentioned ISGOTT (International Safety Guide for Oil Tankers and Terminals) which provides more detailed information on these various sampling instruments and the gases for which they are suitable in Chapter 17.

Special mention is also made of the portable instruments in paragraph 3.14.12 of the Inert Gas Guidelines.

To Mr. P. Stanney:

The Author is grateful to Mr. Stanney for his comments and for sharing his direct experience in the fitting and operation of inert gas systems.

Unfortunately the operation of an inert gas plant is not one which can be controlled from plan approval aspects although the various difficulties of first fitting the plant and then trying to operate it without producing excessive soot are appreciated. It is sometimes possible to increase the load on the boiler to give more efficient burning by running ballast and other similar pumps but this, of course, can only be done where steam driven equipment is provided and, in the days of diesel engines rather

than turbine installations, this method of loading a boiler is rapidly disappearing.

If excessive soot cannot be controlled in the burning process then the only way of removing the deleterious effects of the carry over of solids is to design scrubbers to do this and this is a matter for designers rather than for classification. It is also a matter of maintenance and where such problems are known to occur it is up to the ship Owners to arrange their programmes to ensure that, so far as practicable, the defects mentioned by Mr. Stanney are avoided. The Author is of the opinion that, in due course, the deck piping of an inert gas system may well be made of glass re-inforced plastics which would at least eliminate most of the corroded pipework.

Monitoring the uptake gases during the first system survey for 0^2 and solids content could well be done but it is thought that it would be a brave surveyor who said that a boiler, once fitted on board, was not suitable for the production of inert gas. The suitability of the boiler should really be taken into consideration at the design stage but it is appreciated that the final design can be difficult taking into account all the parameters involved.

To Mr. H. J. M. Jonkers:

Mr. Jonkers has raised several interesting points and his comments are appreciated.

Regarding the use of a double block and bleed arrangement instead of a deck water seal, paragraph 5,15,7.5.5 of the Rules (same as Reg. 62.10.1 of SOLAS) does not allow any alternative to a deck seal for crude oil, product and combination carries. However, paragraph 9.1 of Resolution A.567(14) does allow an alternative arrangement in the case of chemical tankers and in the Author's experience of plan approval on several newbuildings this has invariably been a double block and bleed arrangement which is acceptable for class and was also accepted at the time by the National Authorities concerned. However, just because one or two National Authorities accept a particular arrangement it does not mean that all other Authorities are of the same mind and it is always prudent to ensure that the National Authority concerned has also approved the system. In practice there has been no instance, since the 1981 SOLAS Amendments, of a double block and bleed system having been proposed for use on a crude oil or combination carrier although where it has been proposed to call a ship a chemical/product tanker the guiding light is that it is firstly a chemical tanker, in which case Resolution A.567(14) applies, and then the logic is that you would not take off the double block and bleed just to replace it with a deck water seal for when it is carrying products. The logic of rules and regulations in such circumstances sometimes appears to be illogical.

Indication of power failure on the oxygen monitoring equipment is a harder question to answer and I have enlisted the help of Mr. Stanney, one of the contributors, in answering this question.

Control systems can be operated by hydraulic oil, compressed air or electrical supply and equipment, or a mixture of these, and power failure is an indication that the control system, or part of it, has failed and that, as a result, inert gas may not be capable of being supplied to the deck system.

The oxygen analyser is one input into the control system along with various flow, level and temperature switching devices which are normally arranged to "fail safe", that is, if any one is disconnected the system will either alarm or shut down. The switched input from the oxygen analyser should also be arranged to fail safe, that is, be arranged to give an alarm signal if the analyser is not switched on, as well as at the high oxygen content. In the case in question it would appear that the analyser output switch was not arranged to fail safe but this point is not specifically covered by the Rules. One must add perhaps that the control systems vary and points to be taken into consideration are whether any particular switch is off, on standby, in the running position or whether any switches can be over-ridden for any reason. For example, an oxygen analyser may have to be switched on for up to two hours to get a reliable reading before the inert gas system is put into operation.

So far as the oxygen content reading scale or range is concerned it is agreed that the alarm condition of 8% oxygen content should operate whichever scale is being used and alarm both in the engine and cargo control rooms as per paragraph 5,15,7.7.12.

With regard to the power failure alarm being extended to include the indicated pressure alarm of isolated slop tanks on board combination carriers the Author is of the opinion that since the inert gas Rules have just come into line for both IMO and the Society it would cause difficulties for the Society to step out of line again unless it was of vital importance. It is considered that the operational requirements for regular tests of pressures and atmospheres on board combination carriers should ensure that no exceptional dangerous conditions arise.

Mr. Jonkers' opinion that maintenance records together with a spare P.J. valve should be kept on board is a good idea but unfortunately it would be very difficult to enforce.

To Mr. R. M. Hobson:

The Author believes Mr. Hobson has expanded on his comments made at the meeting to include some questions which he is grateful weren't asked at that time since they pose some problems.

Mr. Hobson is quite correct in thinking that a spade blank is inserted whilst a tank is already full of inert gas preparatory to gas freeing before entry. Assuming the inert gas and venting systems are common it is understood that before opening the hatch lid to insert the blank the tank would first be purged with inert gas to reduce the hydrocarbon content to less than 2% by volume by opening up the cargo valves in a tank and expelling the vapours, via the cargo pipe lines, at the loading manifold. After this the hatch lid would be opened and the blank inserted and the gas freeing process completed by fitting portable water driven gas freeing fans over Butterworth openings and gas freeing through the open hatch cover, and any purge pipes, until the oxygen content reached 21% at all parts of the tank before entry.

The above procedures may now be a legacy of the old days because for compliance with the present rules and regulations the expulsion of vapours should be in a vertical direction and at 6 metres above the deck. However, whether or not such arrangements are still acceptable on the basis of previous satisfactory usage has not been put to the test since no recent newbuildings have proposed the use of such blanking arrangements.

Regarding the alternative duties with which the combined high velocity/vacuum valve shown in Fig. 21(b) has to cope the answer is "yes"! In the case of operation whilst venting large quantities of gas during cargo loading the valve will be fairly well open whereas, when coping only with climatic conditions, the valve may virtually remain on its seat lifting only sufficiently to allow any excess gas to squeeze out. The main requirement is that the vent valve should return firmly to its seat after having lifted and, being a deadweight valve, no problems have been reported in this respect.

The "popping" of the vent valves referred to by Mr. Hobson is likely, in the Author's opinion, to be "popping" shut rather than "open" and he is surprised to learn that there are tranquil periods on board tankers since he has been given to understand, obviously from tanker personnel, that there is never a dull moment on such ships.

In the Author's experience of reported defects regarding the accumulation of pressure within cargo tanks the Author has heard of only one case where the liquid in the PV breaker was expelled onto the deck. In this particular case the vent valves must either have been not large enough or, possibly, not operating efficiently enough to relieve the pressure. So far as the

Society is concerned paragraphs 5,15,4.2.7 of the Rules indicates that the area of the vents should be based on a gas evolution factor of 1.25 times the maximum design loading rate.

As mentioned in the paper under paragraph 5,15,7.2.4 discharge rates vary depending on the ship/shore arrangements and location of shore tanks and, in the same way, the loading rates must vary and it is therefore up to the shipyard, owners and vent manufacturers to use their best endeavours to fit the correct size of vents but, in the end, the crew on board the ship must control the situation to ensure that no excessive pressures occur.

In conclusion the Author regrets that he managed to pump oil only halfway up Mount Everest and hopes that Mr. Atkinson was not waiting for it at the top since, no doubt, the oil would have been frozen before achieving that pinnacle of success.

POSTSCRIPT

The Author wishes to take this opportunity of updating one or two points made in the paper which are the result of discussions/decisions made during the 32nd Session of the Sub-Committee on Fire Protection held at IMO in January of this year but which, even now, are not conclusive.

The first decision concerns paragraph 5,15,7.2.2(d) of the paper relating to the flame arresting devices which have to be fitted on the outlets of purge pipes and also on the inlets of the gas freeing fans.

It would now appear that any requirement to fit a flame screen at the inlet to a gas freeing fan will be dropped and, indeed, that under certain conditions such as increasing the exit velocity from purge pipes on non-inerted tankers from 20m/sec to 30m/sec even the flame screen or arrester on the outlet from the purge pipe may be dropped. However these relaxations, if they are eventually approved, will be conditional, in the event of loading and ballasting operations taking place at the same time as gas freeing operations, on gas monitoring at deck level and where concentrations of gas exceeding 30%LEL are detected near the gas freeing inlets either the gas freeing operation must be stopped and all openings, to the tank being gas freed, closed or the loading/ballasting operations discontinued.

In the light of the above it will also be necessary to make amendments to the "Standards" detailed in MSC/Circ.373 and this will be completed at the next sub-committee meeting in January 1988.

The second decision made concerns paragraph 2.1.8 regarding cargo tank vents on chemical tankers (NB. later also accepted for oil tankers) whereby, as a result of fire tests carried out by the German Authorities using ethylene/air mixtures, it was ascertained that a pressure vacuum valve with flame screens at the inlet to the vacuum side and also at the outlet from the vent could satisfactorily pass an endurance burning test provided certain parameters were complied with. The full text is reproduced below.

FROM BCH 16/16

PROPOSAL FOR VENTING ARRANGEMENTS ON BOARD CHEMICAL TANKERS

The following arrangements may be accepted as providing a level of protection equivalent to that requested by SOLAS, regulation II-2/59.1.5 and contained in the annex to MSC/Circ.373. The arrangement may be used on non-inerted or inerted cargo tanks with vent pipe sizes up to and including 100mm in diameter.

1. Piping systems referred to in chapter 8 of the IBC Code may be provided with the following:

   ,1 pressure/vacuum valves with the following characteristics:

   — constructed in accordance with the appropriate sections of MSC/Circ.373 excluding the type test procedures for flame arresting capabilities;

   — opening over pressure of at least 0.18 bar;

   ,2 a flame-arresting device fitted at the pipe outlet which has been tested for flashback in accordance with paragraph 3.2.2, and where applicable, paragraph 2.5.4 of MSC/Circ.373;

   ,3 The vacuum side of the valve to be protected with a flamescreen as defined in MSC/Circ.373 or a device tested for flashback.

2. It should be ensured that when the valve is open a minimum flow velocity of at least 10m/sec is maintained at the outlet of the assembly above the flame-arresting device.

However, until such matters are finalised via the Maritime Safety Committee and the results published the venting arrangements will have to be considered on the basis of each ship taking into account all the factors involved.

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STERNTUBE BEARINGS: PERFORMANCE CHARACTERISTICS AND INFLUENCE UPON SHAFTING BEHAVIOUR

R. W. Jakeman

Paper No. 2. Session 1986–87

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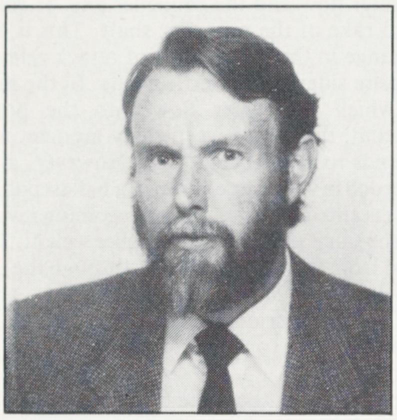
Written contributions to the discussion of this paper are invited from members of the Lloyd's Register Technical Association. To ensure inclusion in the discussion paper, the contributions should be received by the Hon. Secretary in London not later than the 31st January, 1987.

Hon. Sec. C. M. Magill
71 Fenchurch Street, London, EC3M 4BS

STERNTUBE BEARINGS: PERFORMANCE CHARACTERISTICS AND INFLUENCE UPON SHAFTING BEHAVIOUR

by

R. W. Jakeman



R. W. Jakeman served his apprenticeship at the College of Aeronautics, Cranfield, gained a B.Sc. degree from the University of Durham and an M.Sc. degree from the University of Newcastle upon Tyne. His design experience in marine steam turbines and gearing was obtained at Pametrada followed by research and development work on hydrodynamic bearings at Michell Bearings Ltd. and work on the ARG boiler at Clark Chapman & Co. Ltd.

He joined Lloyd's Register in 1971 and, after short initial periods with A.E.S. and MDAPAD, spent 7½ years with T.I.D. He returned to A.E.S. in 1981 and has since been involved in a substantial programme of research on analysis techniques for hydrodynamic journal bearings and in consultancy work concerning shafting alignment and vibration analysis.

SYNOPSIS

Sterntube bearings are introduced with a discussion of the key factors relating to their design and operating environment. The paper gives details of a theoretical study of the performance of oil lubricated sterntube bearings, with respect to both steady and dynamic loading. Particular attention has been given to the effects of angular misalignment. The practical applications of this work to bearing performance analysis and interaction with shaft alignment analysis are described. A comprehensive set of linearised oil film dynamic coefficients, which define the bearing response (forces and moments) to lateral vibration, are presented.

TABLE OF CONTENTS

1. INTRODUCTION

The sterntube bearing is the most critical bearing in relation to the safe operation of any ship. It is generally subject to particularly arduous operating conditions arising from the relatively low shaft speed and close proximity to the propeller. In addition to the consequent deleterious effect on reliability, the sterntube bearing is particularly inaccessible. The result of a total sterntube bearing failure is invariably immobilisation of the ship and the requirement of a drydock in order to carry out repairs. Whilst the effects of dynamic loading combined with angular misalignment are partially mitigated by conservative specific bearing pressures, the reliability of sterntube bearings cannot be regarded as entirely satisfactory.

The subject of this paper forms part of one of the Society's research projects. This project was instigated to improve and expand the facilities available for predicting the static and dynamic behaviour of marine propulsion shafting systems, with the ultimate objective of improved reliability. Particular emphasis was given to the hydrodynamic performance analysis

of bearings subject to angular misalignment; such conditions being usually applicable to sterntube bearings. The area of angular misalignment of bearings in general, has received relatively little attention in published literature.

This paper deals only with bearings in which the running surfaces are fully separated by a fluid film. In relation to current practice, this means that water lubricated bearings are not generally covered, as the low viscosity of water prevents complete hydrodynamic lubrication.

The scope of this paper includes a discussion of design and environmental factors together with consideration of reliability. Following this the theoretical basis of the analysis methods used is outlined, and bearing performance results related to the variation of several design and operating parameters are presented. Finally the practical applications of this work are indicated, with particular reference to the desk top computer bearing analysis programmes that have been developed.

2. DESIGN AND ENVIRONMENTAL FACTORS

2.1 Location

The sterntube, which is an integral part of a ship's aft end structure, forms the passage whereby the propeller shafting passes through the hull. The sterntube bearing (sometimes called a stern bush) is located within the after end of this component and thus supports the weight of the propeller and part of the tailshaft weight. In many cases a bearing is also located at the forward end of the sterntube, but this paper is only concerned with the after sterntube bearing.

In some multiple screw ships with fine hull lines the aftermost part of the propeller shaft may be supported outboard by bearings mounted in "A" brackets or similar structures. In general such bearings are water lubricated and consequently beyond the scope of this paper. Some oil lubricated "A" bracket bearings have been used, and, having regard to the nature of their design and loading, may be treated as sterntube bearings.

2.2 Static Loading

The after end of a propeller shaft is subject to substantial cantilever loading due to the overhung weight of the propeller. This frequently leads to static angular misalignment between the journal and sterntube bearing. Application of rational alignment analysis to the complete propeller shaft system enables such misalignment to be reduced to acceptable levels. The theoretical basis and related measurement techniques for alignment analysis have been described by Archer and Martyn[1].

Shafting design constraints lead to relatively long sterntube bearings, L/D ratios being typically in the region of 2 for oil lubrication. In order to offset their sensitivity to misalignment, sterntube bearing clearance to diameter ratios are relatively large ($C_d/D \simeq 0.002$).

2.3 Dynamic Loading

Marine propellers may produce significant dynamic loading due to their operation in a non-uniform flow of water. In relation to shaft alignment considerations, the propeller dynamic loading may be resolved into a moment applied to the shaft end due to thrust eccentricity and a lateral force due to torque eccentricity. Both the moment and the force vary in magnitude and direction cyclically at propeller blade frequency, the moment being more significant with respect to shaft excitation. In general the above excitation is larger in ships with a high block coefficient, such as tankers and bulk carriers. This is due to the correspondingly greater variation in the velocity of water entering the propeller, which is referred to as the wake field. Notwithstanding these comments, there have been a few notable examples of fine line ships, for example refrigerated cargo, in which substantial wake field effects have been recorded.

For single screw ships holding a straight course, the wake field is virtually symmetrical about the vertical axis passing through the propeller centre. The thrust eccentricty moment is therefore generally dominant in the vertical plane, and the torque eccentricty force is correspondingly dominant in the horizontal direction. The above observations may be invalidated by situations in which there is a significant angle between the mean direction of water flow and the propeller axis of rotation for instance due to rake of the propeller shaft. This is due to the consequent change in the blade angle of attack relative to the water on opposite sides of the vertical axis. In the more usual situation in which the water flow into the propeller is substantially axial, the thrust eccentricity moment acts in the sense that it tends to lift the shaft end. However, a few cases have been recorded in which operation in a ballast condition has resulted in the eccentric thrust moment becoming reversed, and thus acting in the same sense as the propeller weight. This is due to air entrained from the surface passing through the upper part of the propeller. In heavy sea conditions, the above phenomenon may be periodic as a result of the propeller breaking or coming close to the surface as a result of the ship's pitching motion.

When a ship is turning, the wake field becomes asymmetric, and this may result in a much greater horizontal component of eccentric thrust moment. In one notable case which was well documented by Vorus and Gray[2], the above moment when turning with the rudder at about 5° to starboard was sufficient to cause the shaft to be forced against the starboard oil groove of the sterntube bearing and thus precipitate failure.

The foregoing comments are intended to acquaint the reader with the significance and complexity of the dynamic load which may act on a sterntube bearing. It is clearly dangerous to generalise about the form of this loading. For example, it may seem a good idea to place the sterntube bearing oil grooves in the top half of the bearing rather than the conventional 3 o'clock and 9 o'clock positions. However, before proceeding with such an idea, it would be advisable to check that the sterntube bearing is not top loaded as a result of a high eccentric thrust moment. There is substantial evidence to indicate that this situation frequently exists in ships having a high block coefficient.

2.4 Water Lubrication

Water lubricated sterntube bearings are still used in smaller ships, and in many naval ships, where their overall simplicity is advantageous. The low viscosity of water and multiple axial grooves generally used in these bearings prevent effective hydrodynamic lubrication, thus resulting in relatively high wear-down rates. This failure to attain full hydrodynamic lubrication has been confirmed by Leemans and Roode[3]. The axial grooves originated from chamfering of the lignum vitae staves, which were almost exclusively used for such bearings prior to the introduction of modern synthetic materials. These grooves were intended to ensure adequate distribution of water for cooling purposes. More recently proposals have been made that the axial grooves should be omitted from the bottom half of water lubricated bearings made from certain synthetic materials, in order to promote hydrodynamic lubrication. The elimination of grooves from the bottom half would undoubtedly enhance the chances of attaining hydrodynamic lubrication. If, however, due to the low viscosity of water this mode of lubrication is not attained, then overheating is possible as a result of the diminished access of water for cooling. The crucial fact is that the load carrying capacity of a journal bearing is proportional to the effective lubricant viscosity, and the viscosity of water is of the order of one hundredth of that of SAE 30 lubricating oil.

2.5 Oil Lubrication (Boundary and Hydrodynamic)

The problems of high wear down rate associated with water lubrication, and the consequent short bearing life, have been found to outweigh the advantage of simplicity in larger ships. Oil lubricated sterntube bearings have therefore come into widespread use in recent years. In these bearings full hydrodynamic lubrication is generally attained, and only very small amounts of wear down are normally experienced. Wear in such bearings should only occur during starting and stopping, and possibly during low speed operation.

The research which forms the basis of this paper was entirely confined to situations involving complete hydrodynamic lubrication. In practice this means that, apart from the introductory comments given in section 2.4, the paper only deals with oil lubricated sterntube bearings. As indicated above, even oil lubricated sterntube bearings may operate with boundary lubrication at low shaft speeds, particularly where misalignment is present. Boundary lubrication covers the transition region from stationary conditions to full hydrodynamic lubrication. In this region the bearing load is supported partly by direct contact between the journal and bearing surfaces, and partly by hydrodynamic action. Little is known about the acceptability of operation under these conditions, and there is a need for experimental work in this area, preferably at full scale. The following comments are intended as a qualitative guide to the main factors involved.

Under boundary lubrication conditions there will be a substantial increase in the heat generated. This additional heat originates from the area in which journal to bearing contact is occurring. If the conditions are moderate, this may result only in local wear. In more severe conditions, mainly with respect to increased shaft speed, the surface temperature attained as a result of local heating may cause serious degradation of the bearing material and oil properties. This may lead to a catastrophic bearing failure, e.g. wiping of the white metal. The additional heat due to boundary lubrication further compounds the the problem by lowering the mean viscosity of the oil in the bearing, and consequently reducing the proportion of the load supported by hydrodynamic action.

When boundary lubrication occurs under misaligned conditions, the additional heating is confined to a relatively small area at one end of the bearing. For perfectly aligned conditions, however, if boundary lubrication is experienced, it would affect the full length of the bearing. At first sight, this might be construed to indicate that boundary lubrication is a less serious matter under misaligned conditions. Unfortunately the fact is that with misalignment, boundary lubrication may occur at a much higher shaft speed. Under these conditions, although the additional heating is confined to a small area, the higher speed produces a much greater heating intensity.

Present design philisophy is directed towards ensuring that complete hydrodynamic lubrication is attained under all continuous operating conditions. There is, however, much practical evidence to suggest that this is not always achieved. It may be difficult to distinguish between bearing wear originating from starting, stopping and operation on turning gear, and wear incurred during operation in the normal running speed range. Evidence of local overheating would be an indication of the latter, as would wear in the top half of the bearing. From the design viewpoint it may indeed be reasonable, in principle, to accept some degree of boundary lubrication at certain conditions within the running range. However, in the current absence of reliable guidelines for boundary lubrication acceptance criteria, it is considered prudent to aim for hydrodynamic lubrication as far as practicable.

2.6 Oil Feed Grooves

The commonly used oil feed groove arrangement for sterntube bearings is two axial grooves at the 3 o'clock and 9 o'clock positions covering the full, or nearly the full length of the bearing. For the optimum hydrodynamic performance, the location of an axial oil groove should be at the position of maximum film thickness, which is a function of load and misalignment angle. However, in the majority of sterntube bearings the conventional oil groove arrangement appears to be satisfactory. It would be feasible to determine a better oil groove arrangement for any given installation using the hydrodynamic analysis programme described in this paper. This would require the specification of the vertical and horizontal components of misalignment angle experienced by the sterntube bearing under all significant operating conditions. Unfortunately, at present such detailed information on the sterntube bearing operating conditions is rarely available. The previously cited failure described in reference 2 illustrates the hazards of this situation.

2.7 Oil Supply System

In general engineering practice, the lubricating oil supply to hydrodynamic bearings fulfils the secondary function of removal of most of the heat generated. This usually involves positive circulation of oil through the bearing, and the incorporation of an oil cooler in the closed circuit re-circulation system. Sterntube bearings, however, generally operate at fairly low shaft speeds and the heat they produce is consequently small in relation to their size. In addition to this, the close proximity of sea water to the adjacent structure and shaft end provides an effective heat sink. Experience has indicated that heat dissipation in this way is enhanced by maintaining sufficient water in the aft peak tank to cover the sterntube. For many installations, therefore, positive circulation of oil through the sterntube bearing is not necessary. Oil is supplied to the sterntube under pressure from a header tank, the pressure level being set to slightly exceed that of the sea water at shaft level. This arrangement is designed to prevent the ingress of sea water in the event of seal malfunction. The seal is required to have an adequate degree of flexibility in order to tolerate the lateral movement of the shaft arising from the propeller dynamic loading. This flexibility restricts the differential pressure that the seal can withstand, consequently for ships having a large draught range two or more header tanks may be used to maintain the seal differential pressure within acceptable limits. Fig. 1 shows a typical sterntube bearing lubricating oil system. The system shown features two header tanks to cater for a larger draught range, and some degree of oil circulation through the bearing with provision for oil cooling in the circuit.

2.8 Bearing Materials

The remarks in this section are confined to oil lubricated sterntube bearings. By far the most common sterntube bearing material is the tin based white metal with cast iron backing. Although, as noted earlier, the limits of safe operation in the boundary lubrication regime are not known, service experience indicates that white metal is capable of withstanding some degree of such conditions with only minor wear. The other well known virtue of white metal is that it is soft enough for solid particles contaminating the lubricating oil to become embedded in it, and thus rendered virtually harmless. Lastly, in the event of a catastrophic failure (wiping), consequential damage to the tailshaft is rarely serious.

In recent years reinforced resin materials have been used for oil lubricated sterntube bearings. The main advantages claimed for such materials is that their greater flexibility provides an enhanced tolerance to misalignment, and that they are able to operate on a "get you home" basis after a total seal failure.

A substantial amount of research has been carried out on the influence of bearing elasticity on aligned journal bearings. The main direct influence of bearing elasticity is that the shape of the

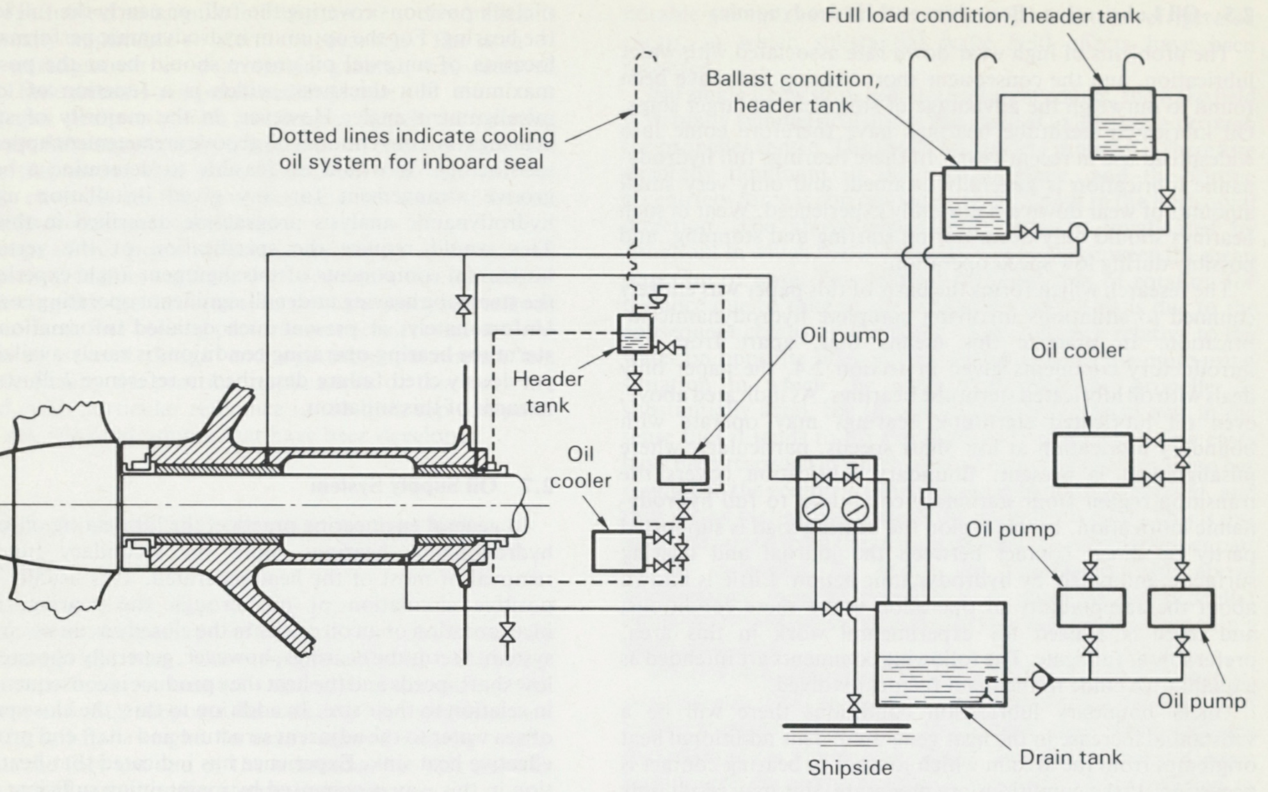


Fig. 1 Typical Arrangement showing Two Header Tanks for Vessels with Large Changes in Draught

bearing surface in the loaded half tends to conform to that of the journal. This is referred to as the "wrap around" effect. The resultant influence on bearing performance depends on the nature of the loading. Fantino et al[4] indicated that where normal hydrodynamic wedge action is dominant, the minimum film thickness is reduced by bearing elasticity. In a more recent paper by La Bouff and Booker[5], the results suggested that minimum film thickness is increased by bearing elasticity in a situation where squeeze film action is dominant. No research results covering the effect of bearing elasticity on misaligned journal bearings, are known to exist at present. The benefits, with respect to misalignment tolerance, of using reinforced resin bearing materials are therefore uncertain, and may well depend upon the severity of dynamic loading.

The main advantage of reinforced resin materials is their ability to operate in water for a limited period. This effectively provides a "get you home" capability in the event of an outboard seal failure. It must be stressed that this is strictly an emergency condition, and that a reduced operating speed is considered to be advisable.

Reinforced resin materials absorb a small amount of water and are consequently subject to slight swelling. The bearing clearance, as machined, must therefore be greater than that for a white metal bearing, in order to allow for swelling.

An important property of reinforced resin materials is thermal conductivity, which is very low relative to white metal. This means that, for monitoring purposes, thermocouples located within reinforced resin materials are virtually useless. The ability of reinforced resin bearings to dissipate heat is very poor. Asbestos is a commonly used reinforcing material, therefore under boundary lubrication conditions, the frictional heat generated may substantially exceed that for white metal. No reliable test data for such bearings is known to exist. Service experience, however, has indicated that under catastrophic failure conditions, severe overheating may occur resulting in serious damage to the tailshaft.

2.9 Reliability

A survey of sterntube bearing and aft seal defect rates has been carried out for the period January 1972 to June 1983. The results were compiled by the Technical Records Department from surveyors reports, and cover over 11,500 ship years of service. These results are summarised in Table 1.

It will be noted that most of the data given is for aft seal and bearing defects combined. This is due to the fact that aft seal defects invariably result in consequential damage to the bearing. It may also be noted that the defect rate for sterntube bearings only is substantially less than the combined seal and bearing defect rate. The apparent conclusion that the seal is a far more critical item than the bearing does not, however, take account of the interdependence of the seal and bearing performance. For the seal, the most exacting operating parameter is the amplitude of lateral shaft movement that it must be able to accommodate. This movement is partially dependent on the dynamic operating characteristics of the sterntube bearing, but the development of predictive techniques has not yet attained a level where the problem can be accurately quantified.

The most interesting result of this survey is the significant direct correlation between defect rate and shaft diameter. In view of the current trend towards the use of larger propellers operating at lower speed, in order to improve propulsive efficiency, there is a clear need to pay more attention to sterntube bearing design and performance. More recently, a limited number of vessels have been fitted with freely rotating vane wheels. The vane wheel is supported on an extension shaft aft of the propeller, with the object of recovering some of the kinetic energy from the propeller wake, which it converts into additional thrust. From the viewpoint of the sterntube bearing, the vane wheel represents a substantial additional mass (up to 50% of the propeller mass) acting at a much greater over-hang distance.

The number of oil lubricated reinforced resin sterntube bearings in service was about 8% of the total, the remainder

Table 1 Failure Statistics

After seal and sterntube bearing defects per 100 ship-years

Diameter (mm)	Defect rate
400–499	4,24
500–599	5,40
600–699	6,08
700–799	6,35
>800	7,03
Overall	5,42

Overall—Sterntube bearing only 1,12 defects per 100 ship-years

Bearing Material	Seal and bearing defect rate
White metal	5,98
Reinforced resin	8,35

No significant correlation of defect rate with:
Ship type Number of propeller blades Number of propellers Fixed or controllable pitch propellers

being almost exclusively white metal. Despite this small proportion, the relative defect rates for these materials were considered to be statistically significant. This data suggests that the previously outlined problems associated with reinforced resin sterntube bearings may in fact outweigh the advantages.

3. THEORETICAL ASPECTS

3.1 Assumptions

In common with theoretical work generally, it is necessary to make various simplifying assumptions in order to reduce the problem to a manageable level. A more complex analysis involving less assumptions is usually possible, but the type of application should be taken into consideration. A highly complex, computationally time consuming, type of analysis may be perfectly acceptable as a research tool but unjustifiable for regular practical application. In practical applications, more approximate but computationally efficient analysis methods are preferable, provided they are backed up by adequate service experience.

Full details of the assumptions made in this work are given in reference 6. In relation to their application to sterntube bearings, the following comments are relevant:

Since sterntube bearings operate at relatively low speed and moderate specific pressure (W/LD), the resultant oil temperature rise is small. In view of this, the commonly used assumption of a constant effective oil viscosity is reasonable. The low speed also ensures that sterntube bearings operate well within the laminar flow region, and that lubricant inertia effects may be neglected.

The modest loading, and the anticipated low level of thermal distortion, justify the assumption of rigid circular journal and bearing surfaces for white metal bearings. For reinforced resin sterntube bearings, the relatively high flexibility of such materials renders the assumption of rigidity unsatisfactory. The application of the Society's current analysis programmes to reinforced resin sterntube bearings inevitably involves some loss of accuracy, and further research is required on the effects of elasticity, particularly for misaligned bearings.

3.2 Hydrodynamic Analysis Method

For comprehensive details of the fundamental hydrodynamic analysis method used in this work, the reader should consult reference 6. The following is a brief outline of the essential features of this method:

In the last hundred years, papers on hydrodynamic analysis have invariably started with Reynold's equation. It is therefore appropriate to explain why reference 6 is an exception to this general rule. At the time it was written (1886), Reynold's equation was incapable of solution, except by means of approximations which yielded results of somewhat questionable accuracy. In addition, whilst this equation was based upon flow continuity within a complete lubricant film, it did not take account of cavitation, to which flow continuity is also generally applicable.

The advent of numerical analysis methods, made practicable by the development of the digital computer, have made it possible to solve Reynold's equation without the former approximations, which were mainly related to bearing length to diameter ratio. Where a numerical analysis method is to be used, there is little value in writing a general partial differential equation, particularly where this does not cover all the conditions encountered, i.e. cavitation. Such an equation must, in any event, be modified to a form suitable for the application of numerical analysis techniques before the solution can proceed. The analysis method used for this work eliminates the initial use of Reynold's equation by going directly to consideration of flow continuity in rectangular oil film elements. In addition to simplifying the analysis process, this approach readily facilitates taking account of continuity in the cavitation zone, which is beyond the scope of Reynolds's equation.

A Gauss-Seidel relaxation method, with successive over relaxation, is used in the above numerical analysis method [6]. The system by which flow continuity is taken into account both within the full film and cavitation regions is believed to be one of the simplest yet developed, and has been proved to function satisfactorily over a wide variety of static and dynamic conditions. In other comparable cavitation models it has been necessary to make initial estimates for the location of the cavitation zone boundary, and to specify the pressure gradient at the rupture boundary. There are no such requirements in the above analysis method. Only specification of the constant cavitation pressure is required, the location of the cavitation zone boundaries being automatically determined to the nearest

rectangular oil film element boundary, during the film pressure relaxation process.

The analysis can handle any geometric condition such as angular misalignment between the journal and bearing axes. This simply requires specification of the geometric conditions such that the oil film thickness can calculated at any oil film element location by means of appropriate trigonometric relationships. Fig. 2 defines the appropriate geometric conditions for a misaligned sterntube bearing. In this figure curvature of the journal axis is neglected.

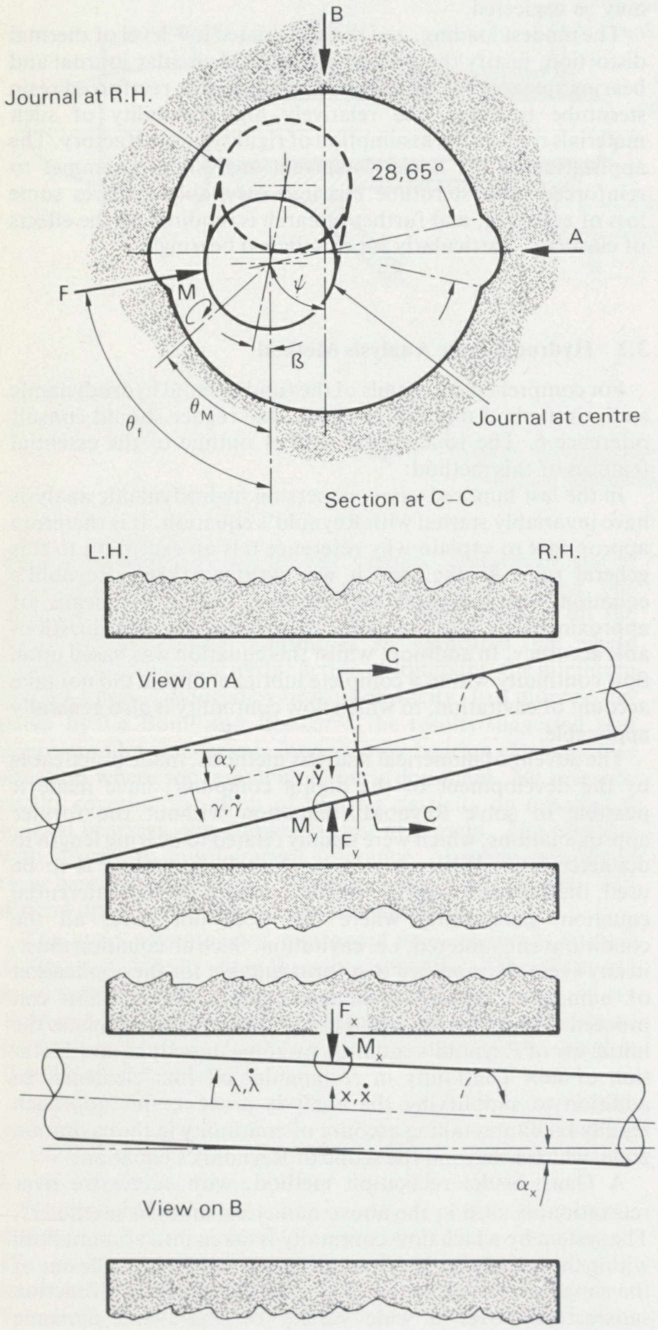


Fig. 2

3.3 Cavitation

In the previous section reference was made to the way in which cavitation has been modelled, by taking account of flow continuity within cavitating elements. This is a substantial improvement on earlier analysis methods in which continuity within the cavitation zone was ignored. The method can be readily incorporated into a film pressure solution where the relaxation technique is used, with little added complexity or computing time. Whilst this method is considered to be quite adequate for most practical applications, it is important to be aware of the approximations in relation to observed cavitation behaviour in real bearings.

The value of the assumed constant cavitation pressure is essentially dependent on the type of cavitation. There are two basic types of cavitation: vaporous and gaseous. For the vaporous type, the pressure within the oil film must drop to the local vapour pressure which, for practical purposes, is virtually zero absolute. Where the oil has absorbed air to saturation level, gaseous cavitation (i.e. air bubbles coming out of solution) may occur at approximately atmospheric pressure. The type of cavitation that is dominant in any particular bearing situation depends on the operating and environmental conditions. Guidance for this can only be very approximate at present, but in most practical bearing situations gaseous cavitation appears to be dominant, thus an atmospheric cavitation pressure is generally appropriate.

Experimental work, such as that by Etsion and Ludwig[7], has given useful insights into cavitation behaviour in bearings, but the bearing geometry and test parameters have not invariably been representative of normal service conditions. Application of such data to practical bearing analysis is therefore difficult. The general conclusion that can be drawn is that the occurrence of gaseous or vaporous cavitation is largely dependent on the time available for bubble release and re-absorption. Vapour release and reabsorption appear to be very rapid in relation to the corresponding times required for gas. Vaporous cavitation is therefore likely to be significant only in dynamic situations affording inadequate time for gas release and re-absorption. Where a bearing is surrounded by air, cavitation zones may be fed with air from the oil film boundaries. This is referred to as ventilation, and clearly cannot occur in fully submerged situations of the sterntube bearing type. The finite release time associated with gaseous cavitation, is believed to be responsible for the frequently observed negative pressure spike preceding the cavitation rupture boundary. Fortunately, this phenomenon appears to have little effect upon load capacity prediction accuracy, except for very lightly loaded bearings. The finite re-absorption time appears in some instances[7] to result in some cavitation bubbles being carried beyond the reformation boundary into the region of increasing film pressure. This clearly results in some variable degree of compressibility in the region of the film where such bubbles persist, with a consequent reduction in load capacity.

No practicable method of theoretically modelling the effects of finite gas release and re-absorption times are known to exist at present.

The flow of oil through the cavitation zone, in the model used for this work, was assumed to take the form of rectangular section streams of full film thickness. This is referred to as the striated model, and has been used in several other theoretical models. An alternative assumption is that the oil becomes fully detached from the bearing surface, and flows through the cavitation zone in a layer adhered to the journal surface. This model was used by Pan[8] and is referred to as the adhered film model. Provided continuity is taken into account within the cavitation zone, the striated and adhered film cavitation models yield the same bearing load capacity for a given effective oil viscosity. The adhered film model, however, results in a lower predicted power loss, since within the cavitation zone the shaft and bearing surfaces are fully separated by air. The power loss is therefore negligible in the cavitation zone. Experimental work such as that by Heshmat and Pinkus[9] has indicated that in reality the form of the oil flow through the cavitation zone may lie between the striated and adhered film

models. This is illustrated in Fig. 3. No practical method of modelling the flow cross section through the cavitation zone more realistically, is known to exist at present. Errors arising from the use of the striated or adhered film models are unlikely to be significant in normal analysis applications.

The foregoing comments indicate that cavitation in journal bearings is a very complex phenomenon. Despite this complexity, the relatively simple theoretical model used appears to be adequate for normal practical applications. Indeed, having regard to the relevant uncertainties in any service bearing, such as the level of solid particle contamination and dissolved gas in the lubricating oil, it is doubtful if a more complex model is practicable.

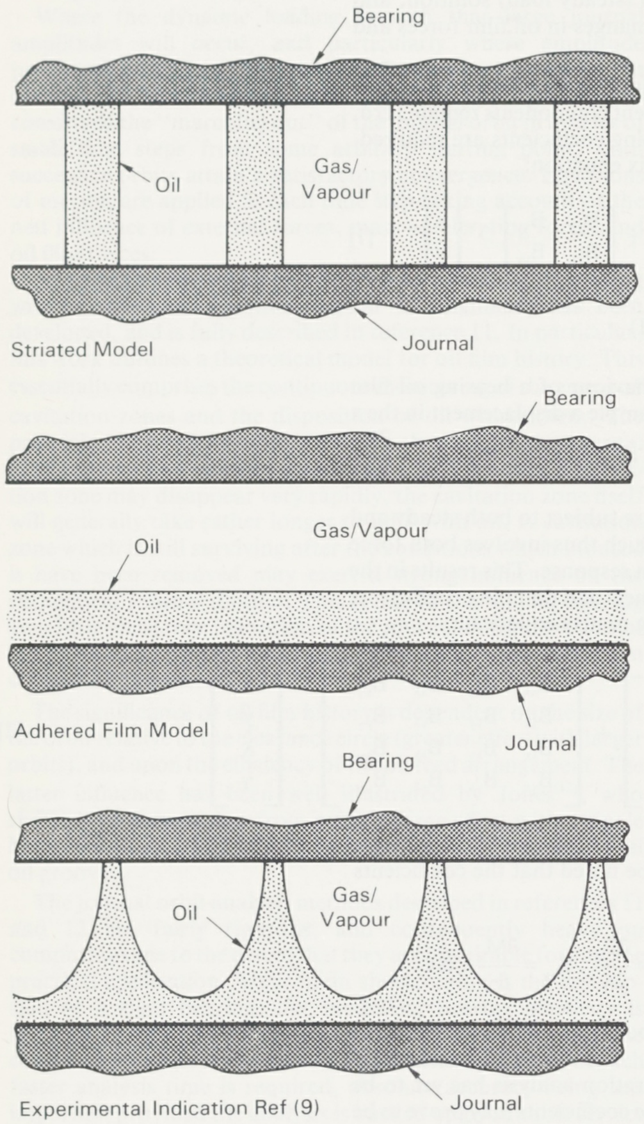


Fig. 3 Axial Cross Section of Flow Through Cavitation Zone

3.4 Element Division

As indicated in 3.2., for the numerical analysis method used, the oil film is divided into a number of rectangular elements. Naturally, a finer element grid will yield a more accurate result, but require more computing time. Clearly some compromise is required, but it is important to note that a finer grid is particularly required where higher rates of change of pressure gradient are encountered. The use of variable element dimensions, (referred to as element grading), therefore offers the possibility of improved accuracy whilst minimising the increase in computing time.

Lloyd et al[10] applied this approach in the circumferential direction, by making the circumferential element dimension a function of the local film thickness. It should be noted that the peak film pressure and maximum rate of change of pressure gradient in the circumferential direction, occur just before the position of minimum film thickness. This method is therefore a simple and effective means of improving accuracy by circumferential element grading.

The above system of circumferential element grading is unsuitable for misaligned sterntube bearings, due to the possibility of substantial variation of the circumferential location of the minimum film thickness position along the length of the bearing. For sterntube bearings the specific pressure (W/LD) is fairly conservative, but substantial angular misalignment is frequently encountered, which results in high local film pressures at one end. In addition, the high L/D ratios of sterntube bearings (typically $L/D \simeq 2.0$) yield a fairly flat axial film pressure profile, which leads to a greater rate of change of pressure gradient at both bearing ends. As a result of the above situations, the maximum rate of change of pressure gradient for a sterntube bearing will often be greater in the axial direction than in the circumferential direction. Consequently there is generally a greater incentive to adopt axial element grading for misaligned sterntube bearings. An axial element grading system was therefore adopted, and a typical element grid is shown in Fig. 4. Note that the circumferential element dimension in the bottom part of the bearing is one half of that for the top part. This effectively introduces a small degree of circumferential element grading within the limitations imposed by misalignment. Where large misalignment angles were encountered, such that significant positive film pressures were likely to be generated in the top half of the bearing at one end, the circumferential element dimension was made the same in both top and bottom halves.

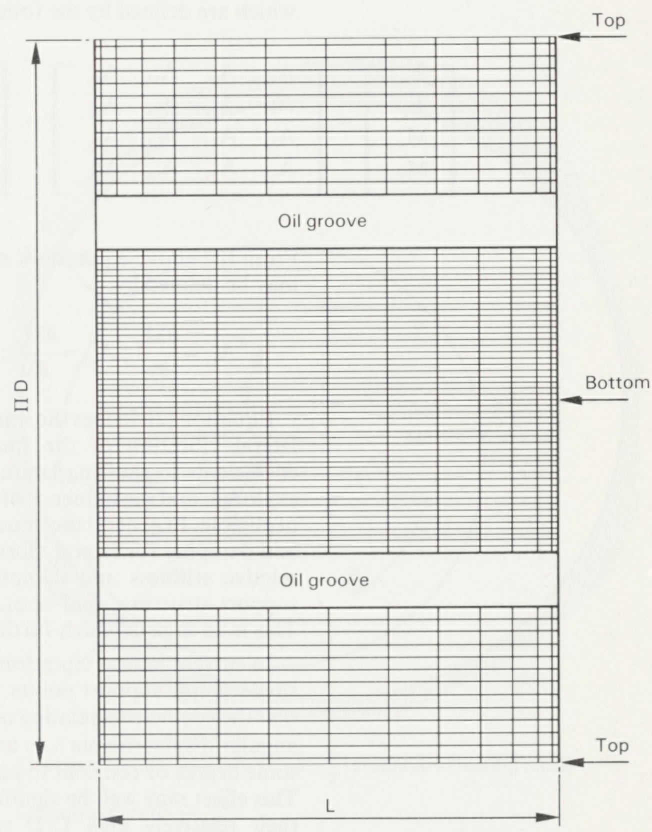


Fig. 4 Oil Film Element Grid

3.5 Oil film Dynamic Coefficients

For dynamic situations in which lateral displacements of the journal are small in relation to the bearing clearance, and the corresponding lateral velocities of the journal axis are also small, the resulting changes of oil film force and moment may be treated as approximately linear. The associated linearised oil film force and moment coefficients are sometimes referred to as displacement and velocity coefficients, but more frequently as stiffness and damping coefficients. The collective term "dynamic coefficient" is also used.

The dynamic coefficients may be computed by superimposing small lateral displacement or velocity changes (referred to as perturbations) on the equilibrium (steady load) solution, and thus deriving the corresponding changes in oil film forces and moments.

In cases where there is no angular misalignment of the shaft and bearing axes, the oil film moment components remain zero, and only eight stiffness and damping coefficients are required. These may be defined by the matrix equation:

$$\begin{bmatrix} F_x \\ F_y \end{bmatrix} = \begin{bmatrix} A_{xx} & A_{xy} \\ A_{yx} & A_{yy} \end{bmatrix} \cdot \begin{bmatrix} x \\ y \end{bmatrix} + \begin{bmatrix} B_{xx} & B_{xy} \\ B_{yx} & B_{yy} \end{bmatrix} \cdot \begin{bmatrix} \dot{x} \\ \dot{y} \end{bmatrix} \qquad [1]$$

(For nomenclature see Appendix I)

It should be noted that the behaviour of a bearing oil film results in cross axis terms. For example a displacement in the x direction results in a change in F_y as well as F_x, thus $A_{yx} = \frac{\partial F_y}{\partial x}$.

In general, sterntube bearings are subject to both steady and dynamic angular misalignment which thus involves both force and moment changes in the oil film response. This results in the need for thirty-two linearised stiffness and damping coefficients which are defined by the following matrix equation:

$$\begin{bmatrix} F_x \\ F_y \\ M_x \\ M_y \end{bmatrix} = \begin{bmatrix} A_{xx} & A_{xy} & A_{x\lambda} & A_{x\gamma} \\ A_{yx} & A_{yy} & A_{y\lambda} & A_{y\gamma} \\ A_{\lambda x} & A_{\lambda y} & A_{\lambda\lambda} & A_{\lambda\gamma} \\ A_{\gamma x} & A_{\gamma y} & A_{\gamma\lambda} & A_{\gamma\gamma} \end{bmatrix} \cdot \begin{bmatrix} x \\ y \\ \lambda \\ \gamma \end{bmatrix} + \begin{bmatrix} B_{xx} & B_{xy} & B_{x\lambda} & B_{x\gamma} \\ B_{yx} & B_{yy} & B_{y\lambda} & B_{y\gamma} \\ B_{\lambda x} & B_{\lambda y} & B_{\lambda\lambda} & B_{\lambda\gamma} \\ B_{\gamma x} & B_{\gamma y} & B_{\gamma\lambda} & B_{\gamma\gamma} \end{bmatrix} \cdot \begin{bmatrix} \dot{x} \\ \dot{y} \\ \dot{\lambda} \\ \dot{\gamma} \end{bmatrix} \qquad [2]$$

From the above equation it may be noted that the coefficients may be defined by:

$$A_{\lambda y} = \frac{\partial M_x}{\partial y}\ ; \ A_{\gamma\lambda} = \frac{\partial M_y}{\partial \lambda}\ ; \ B_{y\gamma} = \frac{\partial F_y}{\partial \dot{\gamma}}\ ; \ B_{\lambda\gamma} = \frac{\partial M_x}{\partial \dot{\gamma}}\ ; \text{ etc.}$$

Equation [2] defines the sterntube bearing oil film response to lateral vibration of the journal. The application of these coefficients to shafting lateral vibration analysis has yet to be explored, and the influence of some coefficients may prove to be negligible. In general the importance of bearing oil film stiffness and damping for lateral vibration analysis, will depend on the relative stiffness and damping of the shafting and bearing support structure, and upon the relative propeller damping. This is an area in which further research is required.

In current lateral vibration analyses, bearings are treated as single simple support points. It is therefore interesting to note that the coefficients relating oil film moment changes M_x, M_y to angular displacements λ, γ and velocities $\dot{\lambda}$, $\dot{\gamma}$ effectively apply some degree of restraint to angular motion of the journal axis. This effect may well be significant in sterntube bearings due to their relatively high L/D ratios. The application of these coefficients thus effectively renders the shaft semi "built in" at the sterntube bearing, rather than simply supported as usually assumed.

3.6 Non-linear Bearing Response: Journal Orbit Analysis

As noted in the preceding section, the application of linearised oil film stiffness and damping coefficients is only valid for small amplitude lateral vibrations. Fortunately, the errors arising from the use of these coefficients in cases where substantial vibration amplitudes are involved are less serious when only natural frequency prediction is required. Little information exists to quantify this situation, but acceptable frequency prediction accuracy has been claimed for cases involving journal amplitudes up to 30% of the bearing clearance. This figure can only be regarded as a very rough guide, since the prediction error will be strongly dependent on the relative stiffness of the shafting and bearing support structure.

Where the dynamic loading is such that large journal amplitudes will occur, and particularly where amplitude prediction is required, linearised coefficients are inadequate. In cases of this type, a journal orbit analysis is appropriate. This consists of the "marching out" of the journal orbit in a series of small time steps from some arbitary starting point, until successive orbits attain a satisfactory convergence. Equations of motion are applied at each time step taking account of the nett influence of external forces, mass accelerating forces and oil film forces.

A journal orbit analysis method for bearings operating without static or dynamic angular misalignment has been developed, and is fully described in reference 11. In particular, this work outlines a theoretical model for oil film history. This essentially comprises the continuous monitoring of the extent of cavitation zones and the disposition of oil within them. The principle effect of oil film history is that whilst the journal displacement and velocity conditions which produce a cavitation zone may disappear very rapidly, the cavitation zone itself will generally take rather longer to refill with oil. A cavitation zone which is still surviving after the conditions which induced it have been removed may exert a strong influence on the subsequent journal motion. This is due to the oil film offering very little resistance to motion of the journal in the direction of the cavitation zone, until that cavitation zone has been completely dissipated.

The significance of oil film history is dependent on the size of the orbit relative to the clearance circle (greater effect with larger orbits), and upon the efficiency of the oil feed arrangement. The latter influence has been well illustrated by Jones[12], who showed the progression from a fairly large effect with a single hole oil feed, to a small effect with a 360° central circumferential oil groove.

The journal orbit analysis methods described in references 11 and 12 are fairly rigorous, and consequently heavy on computing time to the extent that they are unsuitable for routine practical application. Their main shortcoming is the assumption of a rigid bearing, but modelling bearing elasticity is believed to add about another order of magnitude to the computing time (see reference 5). For practical purposes a much faster analysis time is required, absolute accuracy being less important provided the analysis is backed up by suitable service experience.

Many earlier analysis methods used the short bearing approximation of Reynolds equation in order to obtain a fast analytical solution of the oil film force components e.g. Holmes and Craven[13]. This approach can lead to substantial inaccuracy in the oil film force prediction. Cavitation in particular, is simply assumed to extend from the minimum to maximum film thickness positions, this being referred to as the π film model since hydrodynamic action is assumed to occur over an arc of π radians only. Various refinements of the short bearing approximation have been developed, with the aim of improving accuracy. Such methods still suffer from the disadvantages of failing to take account of the oil film boundary conditions arising from various oil feed groove geometries, and of using the very crude π film cavitation model.

The other basic approach to the problem of achieving a fast journal orbit analysis, is to interpolate from a pre-computed or measured data bank of suitable oil film paramenters, in order to obtain rapid force prediction. Probably the best known method of this type is the Mobility Method by Booker[14]. This method has the limitation of only being applicable to bearings having circumferential symmetry e.g. 360° circumferential groove bearings. A fast journal orbit method has been developed by the author, which is based on force prediction by means of five pre-computed velocity coefficients in polar terms:

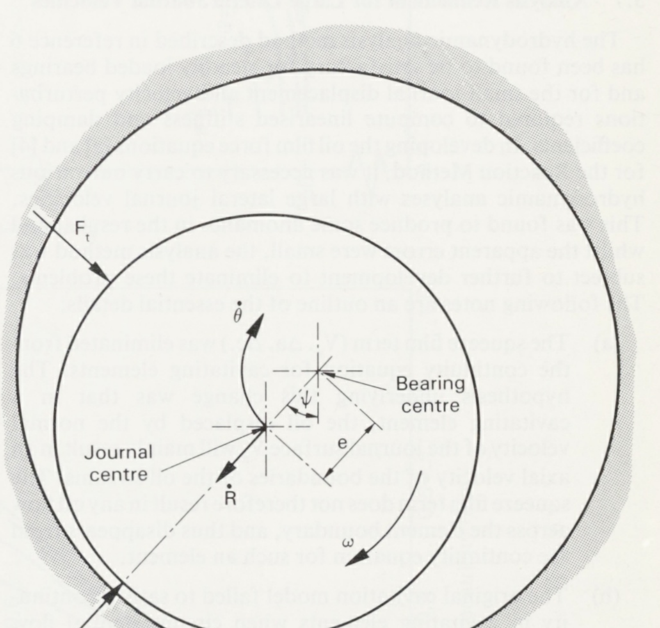
$$F_t = B_{tt}\,\dot{\Theta} + B_{trt}\,\dot{R}\dot{\Theta}_o \qquad [3]$$

$$F_r = B_{rt}\,\dot{\Theta}_o + B_{rr}\,\dot{R} + B_{rrt}\,\dot{R}\dot{\Theta}_o \qquad [4]$$

where $\dot{\Theta}_o = \dot{\Theta} - \omega/2$

The polar velocity and force direction system used is shown in Fig. 5. This analysis method is referred to as the Reaction Method, the development of equations [3] and [4] and application to a 360° circumferential groove bearing being described in references 15 and 16 respectively. The Reaction Method has the advantage of being readily applicable to aligned bearings with any oil groove geometry. As indicated above, this method has been applied to the 360° circumferential groove bearing. The circumferential symmetry of such bearings results in the velocity coefficients B_{tt} etc., of equations [3] and [4] being functions of eccentricity ratio ϵ only. In the more general case of a non-circumferentially symmetrical bearing, the velocity coefficients are functions of both ϵ and attitude angle ψ.

The dynamic excitation from marine propellers is such that the journal amplitude levels in many sterntube bearings may be well into the non-linear area. To explore the shafting behaviour under such conditions, it is proposed to apply journal orbit analysis to the misaligned sterntube bearing. An extensive survey of the literature has revealed no indication of this type of analysis having been attempted on a dynamically misaligned



Effective angular velocity $\dot{\theta}_o = \dot{\theta} - \frac{\omega}{2}$ (stationary bearing case)

Eccentricity ratio $\epsilon = \frac{2e}{C_d}$

Fig. 5 Polar Oil Film—Journal Velocity System

bearing. The nearest relevant work appears to be that by Bannister[17], but this was restricted to a 120° partial arc bearing having an L/D ratio of only 1.0. Furthermore, Bannister only considered static misalignment, i.e. with no cyclic variation of the misalignment angle, and the excitation used was purely out of balance forces with relatively small journal displacement amplitudes. Extension of the journal orbit analysis method described in reference 11 to accommodate dynamic misalignment is not anticipated to pose any special problems. The sterntube bearing situation will clearly be more complex, but the basic analytical approach used for aligned bearings should be adaptable to meet this. It is intended to take account of accelerating forces and moments associated with the propeller and tailshaft mass and inertia, hydrodynamic forces and moments induced by the propeller, and elastic forces and moments from the shafting. The computing time associated with this type of journal orbit analysis will be at least twice that for the aligned condition consequently the work must be initially regarded as purely for research purposes. Application of the Reaction Method, for fast journal orbit analysis, to the dynamically misaligned sterntube bearing may be difficult. The main problem is that the velocity coefficients would become functions of four parameters, namely: eccentricity ratio and attitude angle at the bearing axial centre, misalignment angle and angle of the plane of misalignment. This indicates that a considerable increase in the size of the velocity coefficient data bank would be necessary. In addition, the development of the oil film force equations for an aligned bearing was based on the segregation of squeeze and wedge action, i.e. radial and effective angular velocities. Where misalignment is present, this segregation is impossible due to the variation of attitude angle along the length of the bearing. The above comments do not negate the possibility of applying the Reaction Method to dynamically misaligned bearings, but this will depend on the acceptability of substantial approximations.

3.7 Analysis Refinement for Large Lateral Journal Velocities

The hydrodynamic analysis method described in reference 6 has been found to be satisfactory for steadily loaded bearings and for the small journal displacement and velocity perturbations required to compute linearised stiffness and damping coefficients. In developing the oil film force equations [3] and [4] for the Reaction Method, it was necessary to carry out various hydrodynamic analyses with large lateral journal velocities. This was found to produce some anomalies in the results, and whilst the apparent errors were small, the analysis method was subject to further development to eliminate these problems. The following notes are an outline of the essential details:

(a) The squeeze film term ($V_n . \Delta a. \Delta c.$) was eliminated from the continuity equation for cavitating elements. The hypothesis underlying this change was that in a cavitating element, the oil displaced by the normal velocity of the journal surface V_n will mainly result in an axial velocity of the boundaries of the oil streams. The squeeze film term does not therefore result in any oil flow across the element boundary, and thus disappears from the continuity equation for such an element.

(b) The original cavitation model failed to satisfy continuity in cavitating elements when circumferential flow reversal relative to the h_{min} position occurred; i.e. $\dot{\Theta} < -\omega/2$. Elimination of this problem simply required recognition that, in the above circumstances, element upstream and downstream boundaries became reversed. The essential point is that the gas/vapour flow term computed to satisfy continuity in cavitating elements referred to the downstream element boundary.

(c) The journal surface velocity u was calculated on the basis of the equivalent angular velocity; i.e. $u = (\omega - 2\dot{\Theta})D/2$ instead of the original $u = \omega D/2$. In addition, $\dot{\Theta}$ was deleted from the computation of V_n, therefore V_n became a function of $\dot{R}$ only. The above measures effectively segregated the hydrodynamic squeeze and wedge actions in the analysis. This segregation is unnecessary in full film elements, but is advantageous with cavitating elements. The reason for this is that when eliminating the squeeze film term from the continuity equation for cavitating elements, as indicated in item a., it was found that only that part of ($V_n . \Delta a. \Delta c.$) due to $\dot{R}$ should be eliminated. When applying this analysis modification to a misaligned bearing, it is necessary to apply it individually to each element column, since $\dot{R}$ and $\dot{\Theta}$ will vary with the axial position.

4. RESULTS

4.1 Parameter Study

A comprehensive investigation has been undertaken in which the influence of variation of the following parameters upon the steady load performance and dynamic coefficients was examined: L/D, ϵ, $\bar{\alpha}$, $\bar{p}_h$, β, Θ_f, $\bar{B}_{ow}$.

Exhaustive coverage of a large range of combinations of these parameters was not feasible due to the amount of computation involved. Variation of each parameter singly was, therefore, carried out while maintaining the remainder constant at the following datum conditions:

$L/D = 2.0$, $\epsilon = 0.7$, $\bar{\alpha} = 0$ and 0.2, $\bar{p}_h = 0$, $\beta = 0$, $\Theta_f = 0$, $\bar{B}_{ow} = 0$.

It should be noted that the above treatment is necessary when generating analytical data in this field. This is not representative of the way in which a real bearing behaves, since it is generally impossible, particularly with a service bearing, to change one variable without affecting others. For example, in this type of analysis when $\bar{\alpha}$ was increased ϵ was held constant. Due to the non-linearity of the oil film, when increasing $\bar{\alpha}$ in a real bearing, the corresponding increase in oil film force at the bearing end where the journal eccentricity increases, exceeds the decrease in oil film force at the opposite end of the bearing. The result is that ϵ will decrease in order to maintain the same total oil film force. A practical bearing analysis technique, which utilizes precomputed data of the type presented in this section, is described in section 5.

Computation of the steady load solution only required about 10% of the computing time necessary for a full analysis including all 32 dynamic coefficients. Accordingly a more extensive analysis programme was conducted for steady load conditions, the parameter ranges being:

$L/D = 1$ to 3, $\epsilon = 0.1$ to 0.95, $\bar{\alpha} = 0$ to 0.9, $\bar{p}_h = 0$ to 0.2 with β, Θ_f and $\bar{B}_{ow}$ held at zero.

A few of the more interesting results of this parameter study are presented in 4.2 and 4.3. Complete details of this work are given in reference 18.

4.2 Steady Load Performance

The most significant "independent" variables are ϵ and $\bar{\alpha}$, and the performance parameters of greatest interest are $\bar{W}$, $\bar{H}$, $\bar{Q}$ and $\bar{M}$. The computed relationship between these parameters is illustrated in Figs. 6, 7, 8 and 9 respectively. A relationship clearly exists between $\bar{\alpha}$ and the maximum possible value of ϵ, which is due to the attainment of contact between the journal and bearing at one end. This is reflected in the corresponding limits to the extent of the curves in Fig. 6 to 9. All the performance parameters are highly dependent upon ϵ, but the above figures show that only $\bar{M}$ is substantially affected by $\bar{\alpha}$.

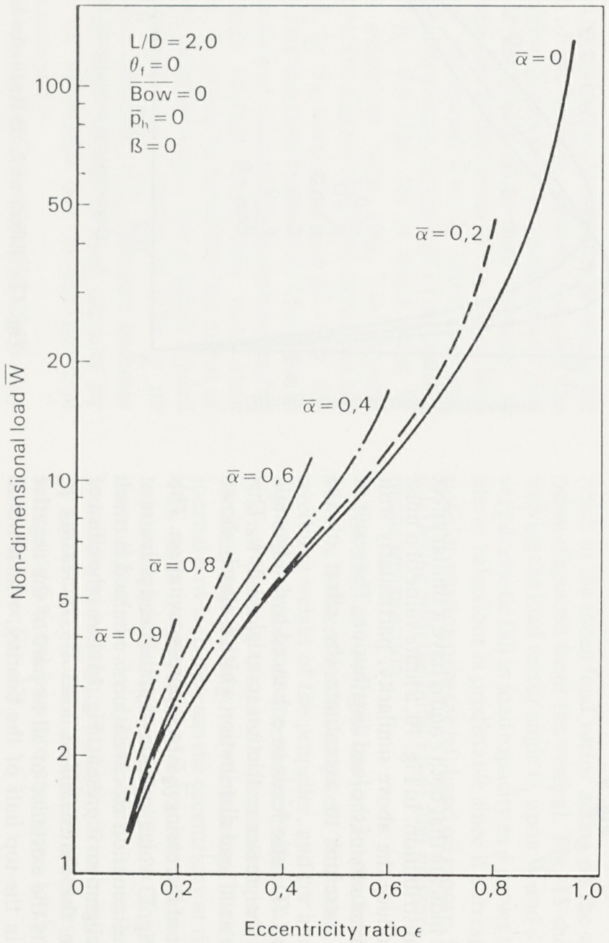


Fig. 6

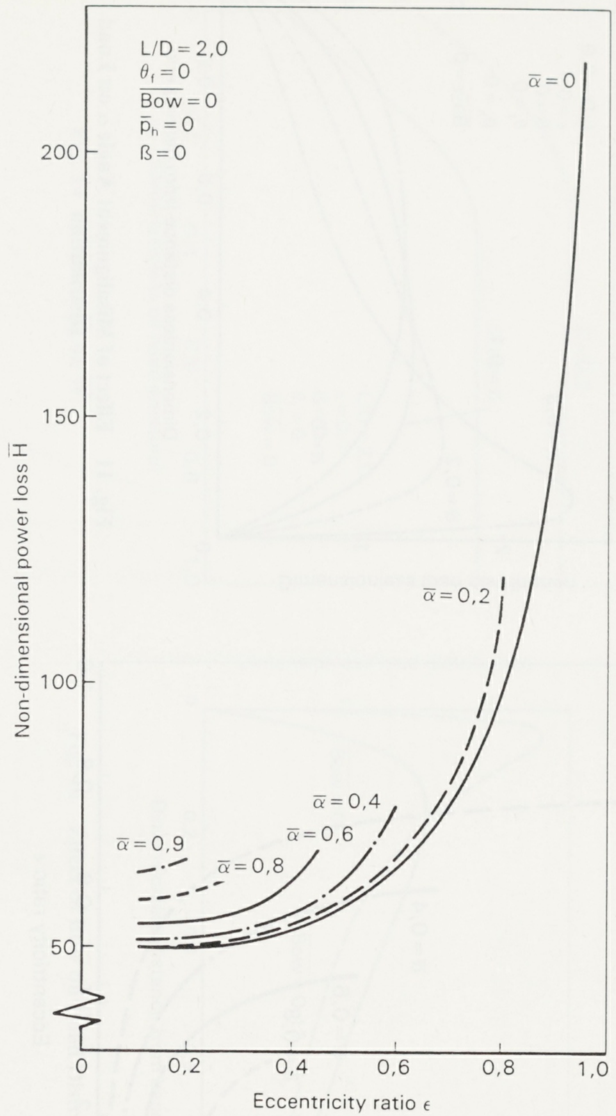


Fig. 7

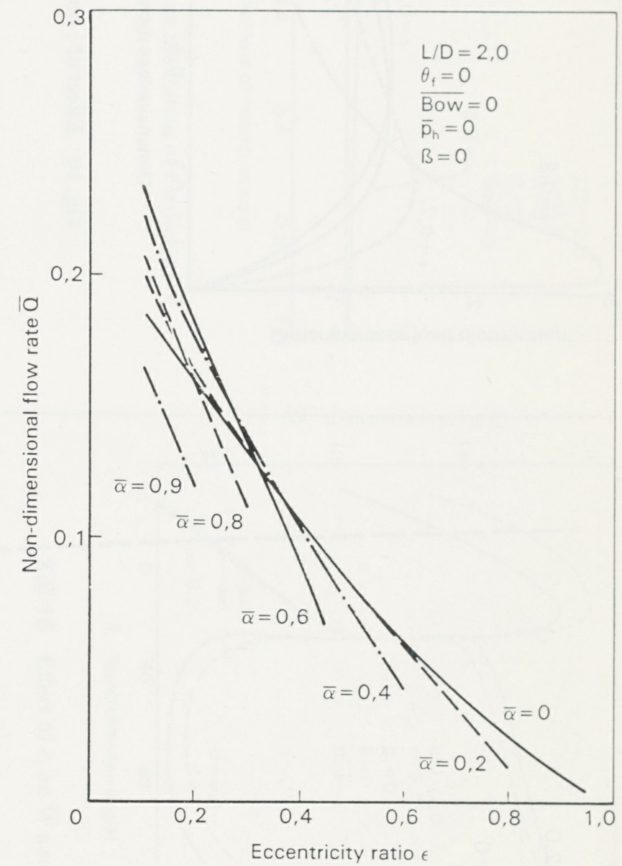


Fig. 8

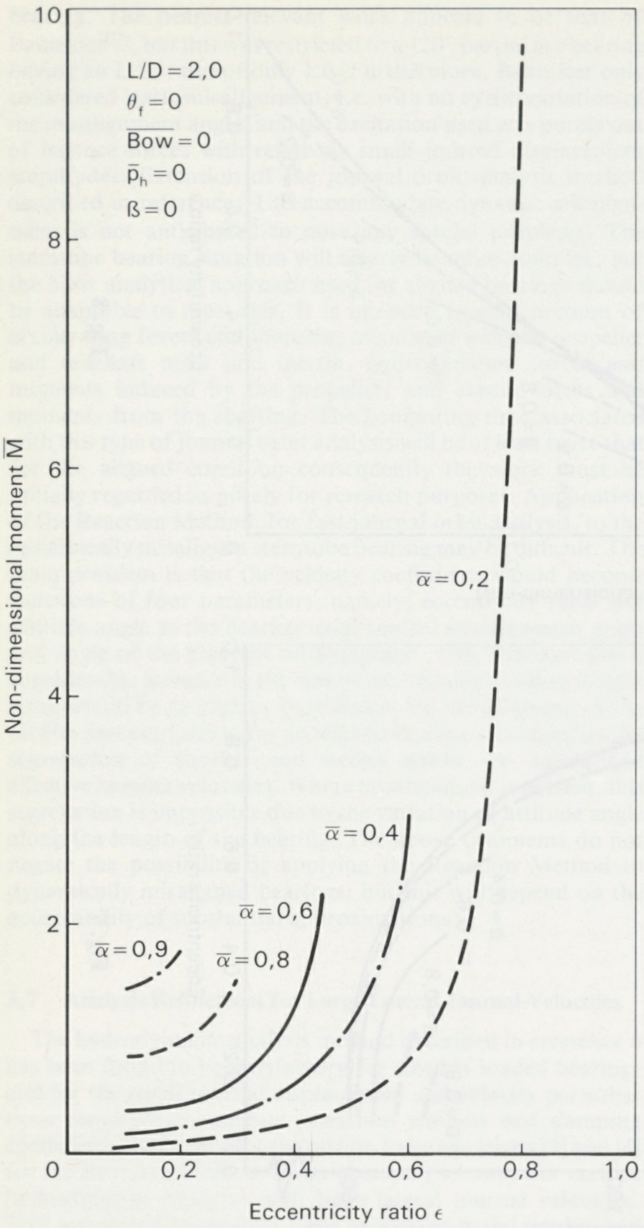


Fig. 9

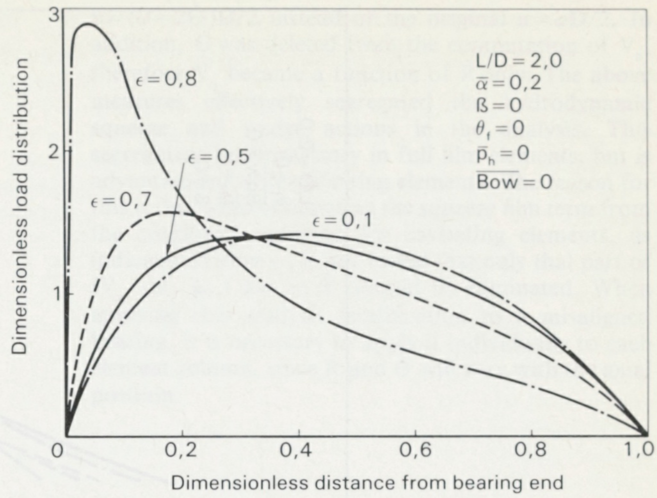


Fig. 10 Effect of ϵ on Load Distribution

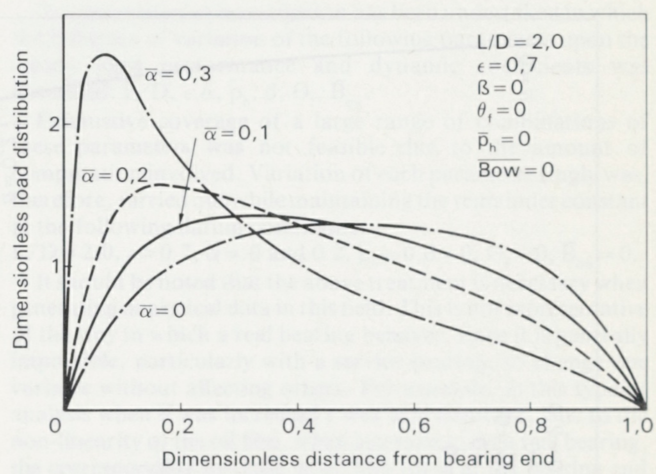


Fig. 11 Effect of Misalignment Angle $\bar{\alpha}$ on Load Distribution

Figs. 10 and 11 indicate that both ϵ and $\bar{\alpha}$ have a similar effect on the axial load distribution. In Fig. 10, a finite value of $\bar{\alpha}$ must be present to produce the above similarity, particularly with respect to the assymmetry of the load distribution. The result of increasing ϵ is therefore to accentuate the effect of the misalignment.

An increase in L/D ratio leads to enhanced hydrodynamic efficiency due to the greater restriction to axial oil flow. This results in a flatter axial load distribution which is clearly shown by Fig. 12.

The main effect of increasing $\bar{p}_h$ is to suppress cavitation. This is illustrated by Fig. 13 which indicates that the head pressure at which complete elimination of cavitation is attained is much higher when misalignment is present. Fig. 14 shows the effect of increasing $\bar{p}_h$ on the dimensionless load $\overline{W}$. Increasing $\bar{p}_h$ effectively permits the existance of all or part of the negative pressure region in the top half of the bearing, which would

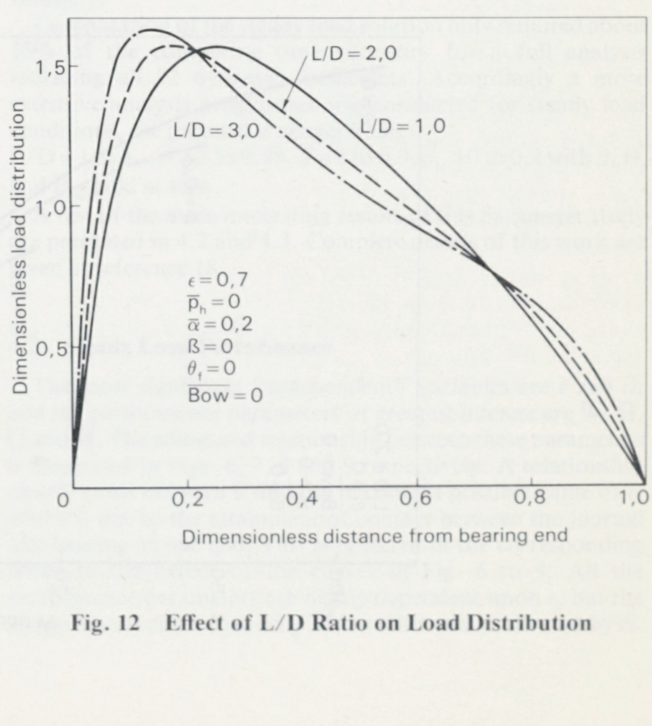


Fig. 12 Effect of L/D Ratio on Load Distribution

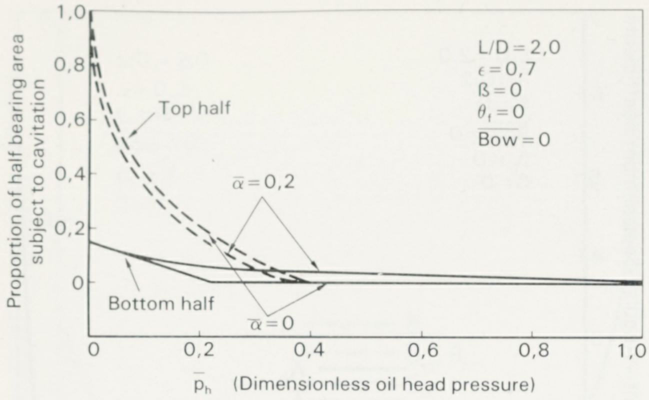


Fig. 13 Effect of $\overline{p}_h$ on the Extent of Cavitation

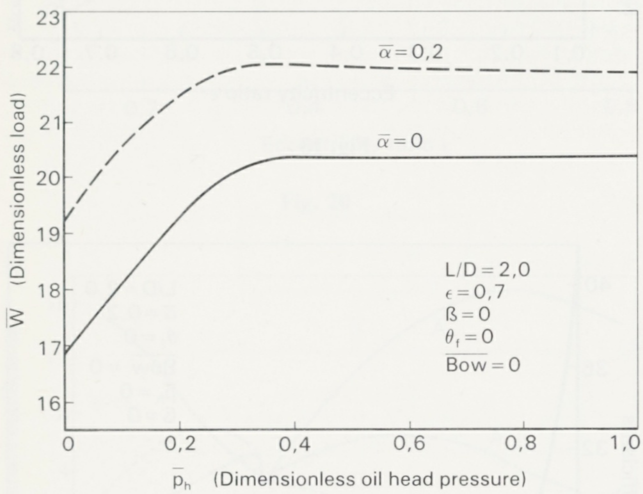


Fig. 14 Effect of $\overline{p}_h$ on $\overline{W}$

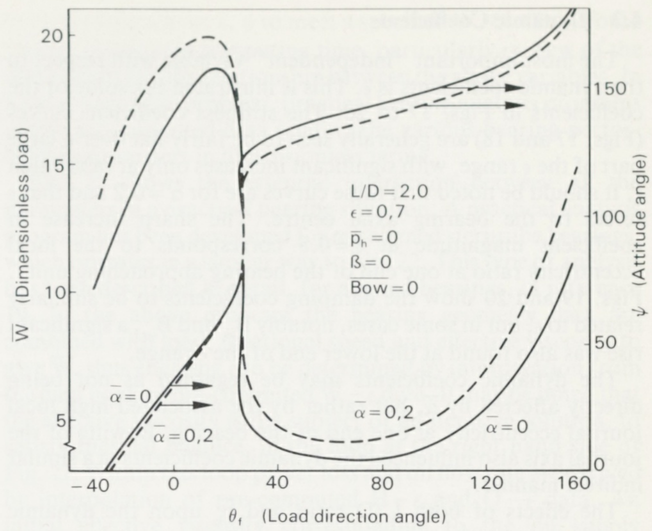


Fig. 15 Effect of θ_f on $\overline{W}$ and ψ

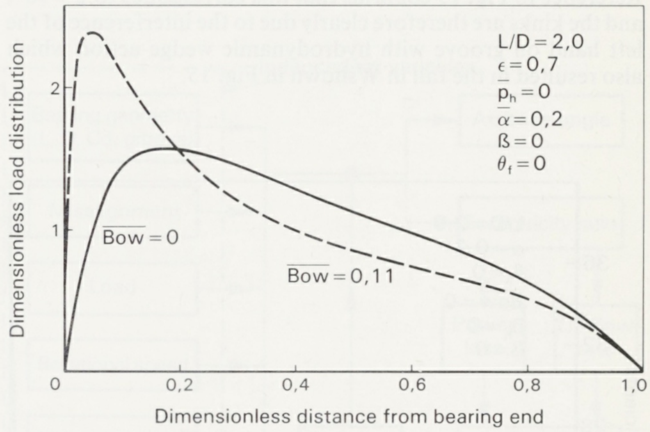


Fig. 16 Effect of $\overline{Bow}$ on Load Distribution

otherwise be eliminated by cavitation. It does this by raising the general pressure level of the oil film, thereby reducing the tendency for the cavitation pressure to be encountered. This results in a significant improvement in load capacity, as indicated by the left hand side of the curves in Fig. 14. In both the aligned ($\overline{\alpha}=0$) and misaligned $\overline{\alpha}=0.2$) cases analysed, complete elimination of cavitation in the top half of the bearing was attained at about $\overline{p}_h=0.4$ (Fig. 13) at which point $\overline{W}$ therefore attained its maximum value (Fig. 14). Beyond $\overline{p}_h=0.4$, $\overline{W}$ is constant for the aligned case since cavitation has been totally eliminated. In the misaligned case ($\overline{\alpha}=0.2$), the value of $\overline{W}$ falls slightly as $\overline{p}_h$ increases beyond 0.4. This is due to the more gradual elimination of cavitation in the bottom half of the bearing at the high journal eccentricity end, when misalignment is present. It should be noted that the extent of cavitation is particularly dependent on ϵ. For a lightly loaded bearing, the correspondingly low ϵ results in little or no cavitation, and the above influence of $\overline{p}_h$ therefore becomes negligible. Unfortunately improvement in load capacity by increasing $\overline{p}_h$ cannot be advocated due to the practical constraints outlined in section 2.7.

As outlined in section 2.3, propeller wake field effects may result in the mean load vector acting on the sterntube bearing being displaced from the vertical. Fig. 15 shows the effect of varying the load vector angle Θ_f upon $\overline{W}$ and ψ. This indicates a very dramatic fall in load capacity as ψ passes through 90°. The above behaviour is predictable since it corresponds to the h_{min} position passing over the left hand oil groove, thereby seriously interfering with the build up of film pressure due to hydrodynamic wedge action.

The effect of bowing of the journal due to the application of bending moment was investigated. This arises mainly from the overhung weight of the propeller, and for analytical purposes has been treated as a constant radius of curvature i.e. the journal bending moment was assumed to be approximately constant over the length of the bearing. The dominant effect of journal bow is to increase the eccentricity at the bearings ends. This tends to improve hydrodynamic efficiency by restricting axial oil flow in the high pressure region. The main results are a flattening of the axial load distribution, and an increased sensitivity to misalignment, both effects being clearly illustrated by Fig. 16. It is evident from the above results that the effects of increasing journal bow are analogous to those of increasing L/D ratio (see Fig. 12).

4.3 Dynamic Coefficients

The most important "independent" variable with respect to the dynamic coefficients is ϵ. This is illustrated for some of the coefficients in Figs. 17 to 20. The stiffness coefficient curves (Figs. 17 and 18) are generally seen to be fairly flat over a large part of the ϵ range, with significant increases only at fairly high ϵ. It should be noted that these curves are for $\overline{\alpha} = 0.2$ and that ϵ refers to the bearing axial centre. The sharp increase in coefficient magnitude at $\epsilon = 0.8$ corresponds to the local eccentricity ratio at one end of the bearing approaching unity. Figs. 19 and 20 show the damping coefficients to be similarly related to ϵ, but in some cases, notably B_{yy} and $B_{\gamma\gamma}$, a significant rise was also found at the lower end of the ϵ range.

The dynamic coefficients may be regarded as not being directly affected by $\overline{\alpha}$, but rather by the associated high local journal eccentricity at one end of the bearing. Bowing of the journal axis also influenced the dynamic coefficients in a similar indirect manner.

The effects of both L/D ratio and $\overline{p}_h$ upon the dynamic coefficients (not illustrated) were generally small.

A rather more interesting effect was that of Θ_f which is shown for the lateral stiffness coefficients (force—lateral displacement) in Fig. 21. The behaviour of the other dynamic coefficients was similar in that distinct kinks occurred at about $\Theta_f = 37°$. Reference to Fig. 15 confirms that this corresponds to $\psi = 90°$, and the kinks are therefore clearly due to the interference of the left hand oil groove with hydrodynamic wedge action which also resulted in the fall in $\overline{W}$ shown in Fig. 15.

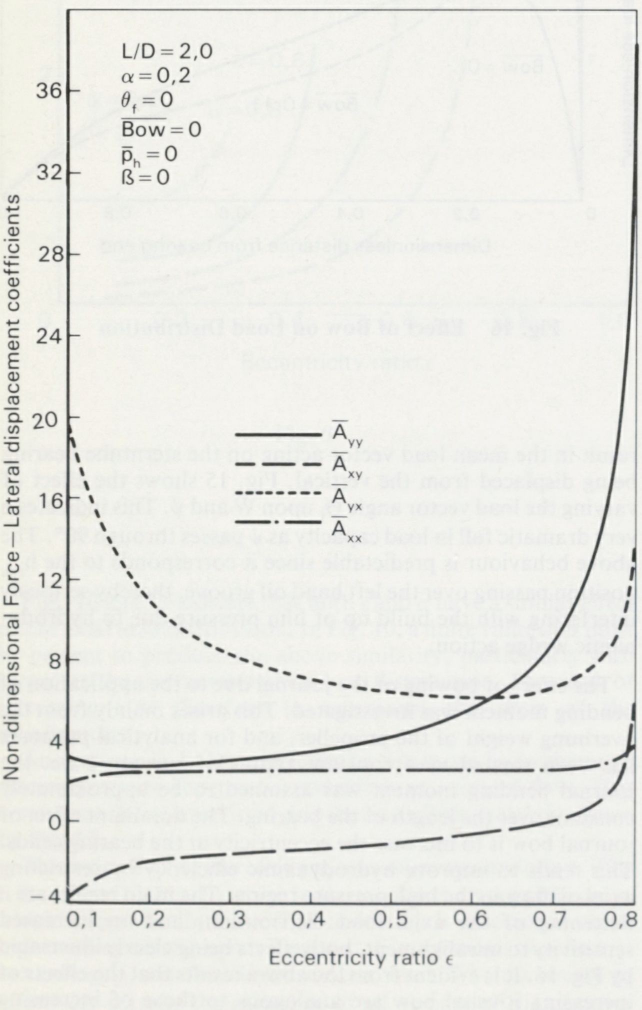


Fig. 17

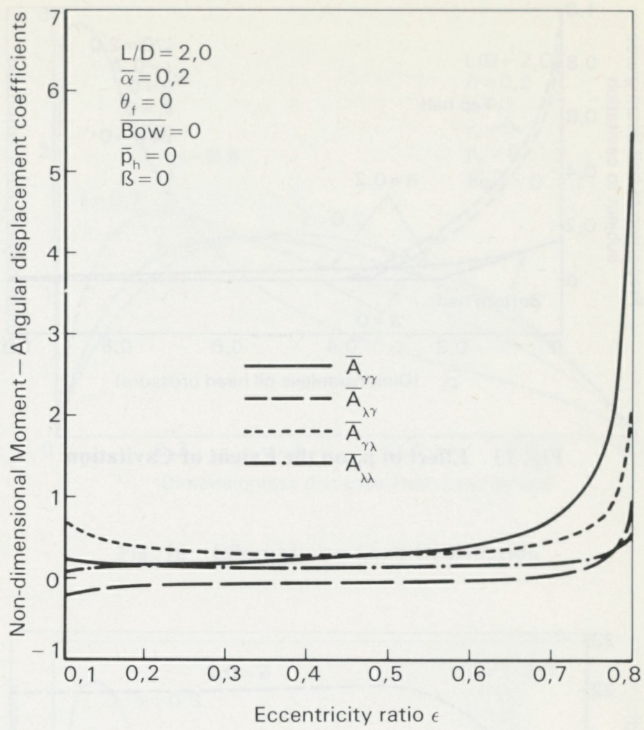


Fig. 18

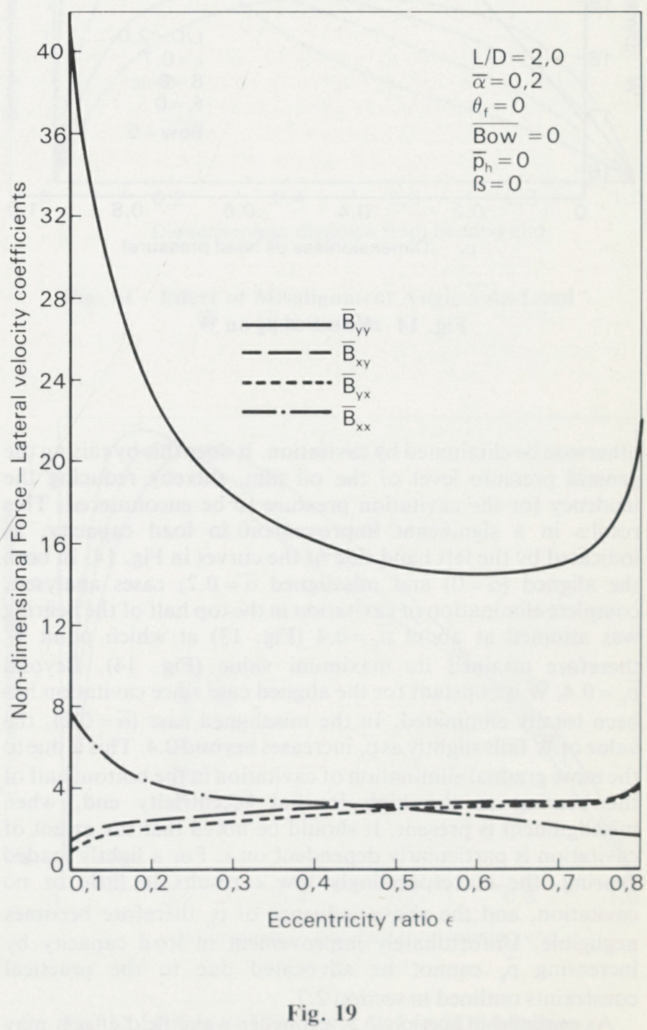


Fig. 19

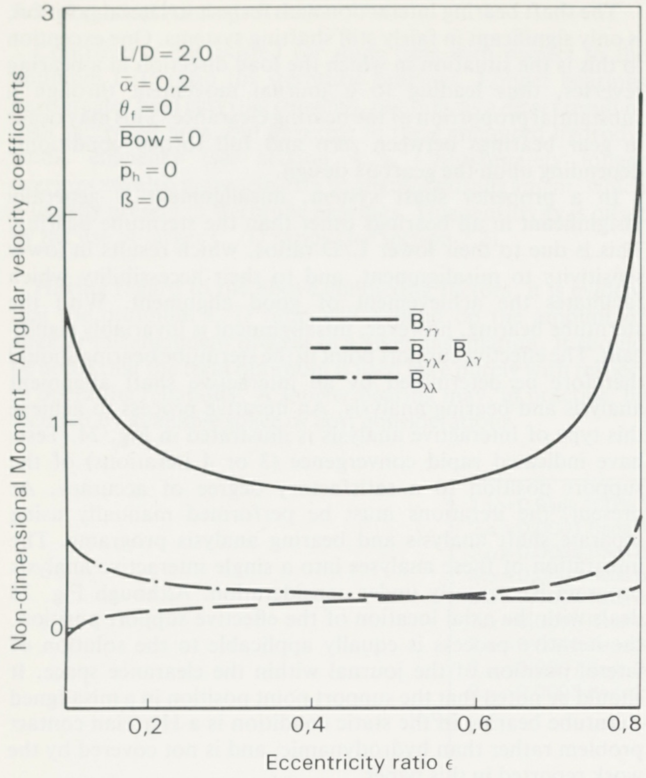


Fig. 20

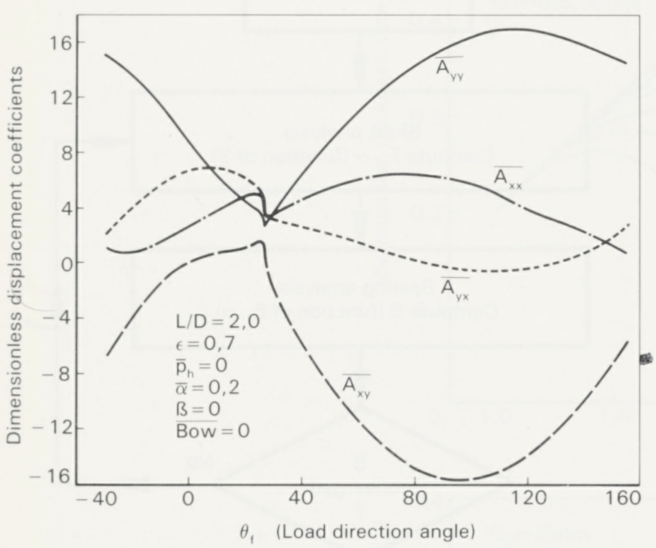


Fig. 21 Effect of θ_f on $\overline{A_{yy}}$, $\overline{A_{yx}}$, $\overline{A_{xy}}$, $\overline{A_{xx}}$, ($\bar{\alpha} = 0,2$)

5. PRACTICAL BEARING ANALYSIS

5.1 Real Bearing Parameter Interaction

An inherent problem of the hydrodynamic analysis method described in section 3.2, is that the position of the journal within the bearing clearance must be specified and the corresponding oil film force computed, i.e. ϵ, ψ are independent variables and W, Θ_f are dependent variables. In the majority of practical applications it is necessary to treat the relationship between the above variables in the reverse sense. The only way in which this can be achieved with a hydrodynamic analysis, is to use an iterative solution for ϵ, ψ to meet a specified W, Θ_f. This would require substantial computing time, particularly in view of the non-linearity of the relationship between the above variables. In view of the above problem, practical bearing analysis programs utilize pre-computed data relating the various bearing performance parameters in dimensionless terms.

Fig. 22 shows the essential interaction between all the parameters in a practical bearing situation. A practical analysis program has been developed for misaligned sterntube bearings, which operates in a similar way to Fig. 22. This type of analysis has been described in detail, for aligned bearings, in reference 19. In the above program the bearing geometry data are combined with load, rotational speed and effective viscosity to give $\overline{W}$, thus enabling ϵ to be determined by interpolation from the data bank of pre-computed $\overline{W} - \epsilon$ data. It may be noted that the effective viscosity is the subject of an iterative solution, to satisfy thermal balance, in the loop at the right hand side of Fig. 22. Within this loop power loss and oil flow rate are derived by interpolation of pre-computed $\overline{H} - \epsilon$ and $\overline{Q} - \epsilon$ data. An initial effective viscosity corresponding to the oil supply temperature is assumed. The iterative thermal balance solution is thus analogous to a thermal transient condition. On completion of the thermal balance solution, ψ, $\overline{M}$ and Θ_m may be interpolated from the relevant data banks to complete the analysis.

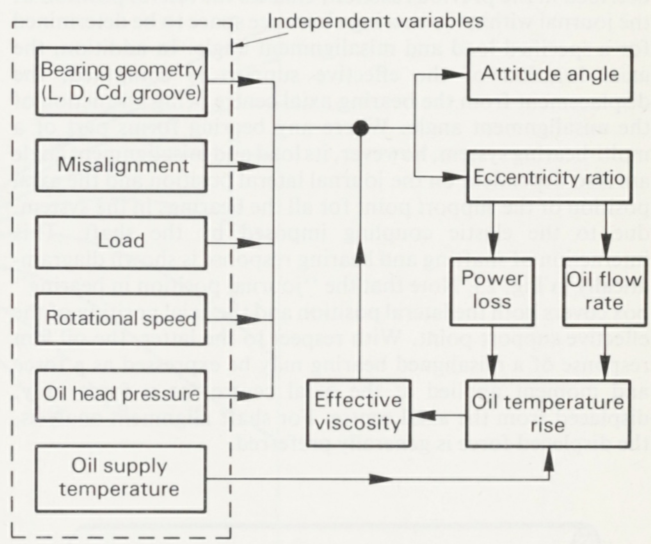


Fig. 22 Bearing Performance: Interaction of Parameters

In the misaligned sterntube bearing program data banks are required for $\overline{W}$, ψ, $\overline{H}$, $\overline{Q}$, $\overline{M}$ and Θ_m. Data for each of these parameters is stored as function of ϵ, $\bar{\alpha}$, L/D and $\bar{p}_h$, and four dimensional matrices are therefore used for this purpose. It follows from the above that the complete data bank comprises six four dimensional matrices, these comprising a total of 4,320 data items. In the present version of the program, variation of Θ_f, β and B_{ow} is not covered, these parameters being assumed to be zero. Expansion of the program to include variation of Θ_f, β and B_{ow} is theoretically possible, but would require an increase in the size of the data bank by about an order of magnitude for each variable.

At present, the misaligned sterntube bearing program is available on disc for use with the Hewlett Packard 9836c desk top computer, the program name being "STBPER". (Stern Tube Bearing Performance). Approximate operating times are as follows:

Load Program:		5 secs.
Initialize Program (reads data bank into operating matrices):		14 secs.
For one analysis case:	Complete Data Input	40 secs.
	Analysis	26 secs.
	Print Out Results	15 secs.

The program has facilities for modifying the input data and re-running the analysis thus reducing the data input time for subsequent runs.

Program "STBPER" does not include computation of the thirty-two oil film stiffness and damping coefficients, since this would require an increase in the number of four dimensional data matrices from six to thirty-eight. The stiffness and damping coefficients may therefore be computed by program "STBSDC". (Stern Tube Bearing Stiffness and Damping Coefficients). This program uses a hydrodynamic analysis model to compute each coefficient, and consequently requires 5 to 6 hours running time on the Hewlett Packard 9836c.

Background notes and operating instructions for programs "STBPER" and "STBSDC" are given in reference 20.

5.2 Bearing—Shafting System Interaction

The misaligned sterntube bearing program (STBPER), decribed in the previous section, enables the lateral position of the journal within the bearing clearance space to be determined for a specified load and misalignment angle. In addition, the axial location of the effective support is computed, the displacement from the bearing axial centre being a function of the misalignment angle. Where any bearing forms part of a multi-bearing system, however, its load and misalignment angle are also dependent on the journal lateral position and the axial position of the support point for all the bearings in the system, due to the elastic coupling imposed by the shaft. This interaction of shafting and bearing response is shown diagrammatically in Fig. 23. Note that the "journal position in bearing" box covers both the lateral position and the axial position of the effective support point. With respect to the latter, the oil film response of a misaligned bearing may be expressed as a force and moment applied at the axial centre, or a force only, displaced from the axial centre. For shaft alignment analysis, the displaced force is generally preferred.

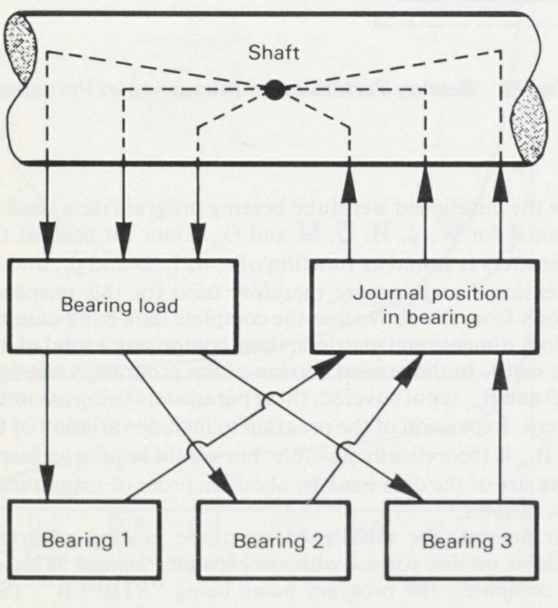


Fig. 23 Bearing Shaft Interaction

The shaft bearing interaction with respect to lateral position, is only significant in fairly stiff shafting systems. One exception to this is the situation in which the load direction in a bearing reverses, thus leading to a journal movement through a substantial proportion of the bearing clearance. This may occur in gear bearings between zero and full torque conditions, depending upon the gearbox design.

In a propeller shaft system, misalignment is generally insignificant in all bearings other than the sterntube bearing. This is due to their lower L/D ratios, which results in lower sensitivity to misalignment, and to their accessibility which facilitates the achievement of good alignment. With the sterntube bearing, however, misalignment is invariably significant. The effective support point in the sterntube bearing should therefore be determined by an interactive shaft alignment analysis and bearing analysis. An iterative process to achieve this type of interactive analysis is illustrated in Fig. 24. Tests have indicated rapid convergence (3 or 4 iterations) of the support position to a satisfactory degree of accuracy. At present, the iterations must be performed manually using separate shaft analysis and bearing analysis programs. The integration of these analyses into a single interactive analysis program is currently under consideration. Although Fig. 24 deals with the axial location of the effective support position, the iterative process is equally applicable to the solution of lateral position of the journal within the clearance space. It should be noted that the support point position in a misaligned sterntube bearing in the static condition is a Hertzian contact problem rather than hydrodynamic, and is not covered by the work reported in this paper.

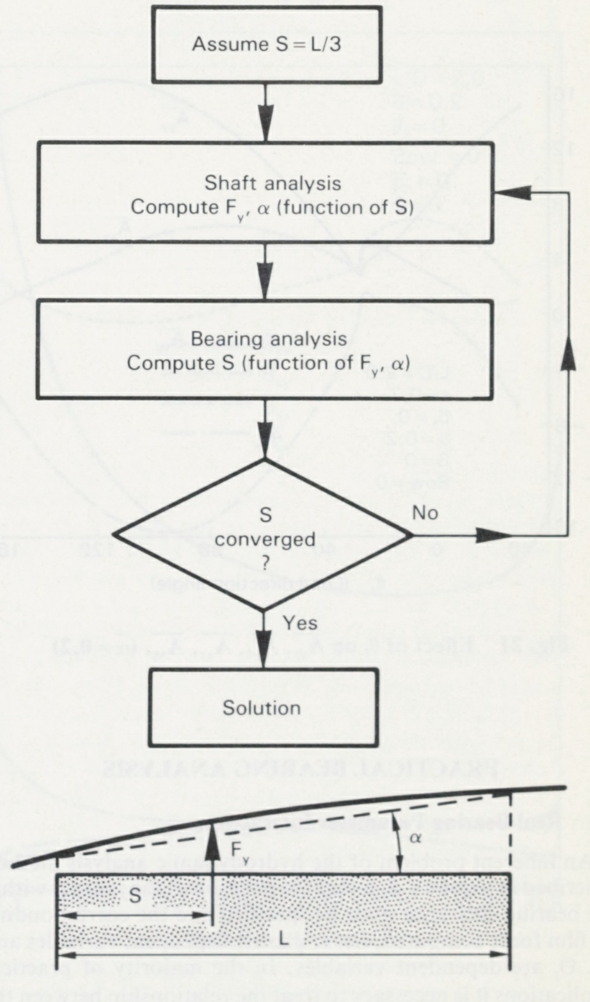


Fig. 24 Interactive Solution for Misaligned Bearing Support Point

5.3 Practical Analysis Results

Controversy has existed for some time with respect to the optimum L/D ratio for sterntube bearings. Longer bearings have greater load carrying capacity, not only as a direct result of their increased area, but also due to their enhanced hydrodynamic efficiency (*see* section 4.2). Unfortunately longer bearings are also more sensitive to angular misalignment, which is usually present in sterntube bearings. This sensitivity is offset by the use of larger C_d/D ratios (*see* section 2.2).

Having regard to the above conflicting factors, it is evident that the optimum L/D ratio will depend on the amount of angular misalignment present in a given installation. It should be noted that the term optimum is used here purely with respect to operating reliability. A shorter bearing may, in practice, be dictated by available space or cost considerations.

The misaligned sterntube bearing program "STBPER" enables the optimum L/D ratio to be determined for any given operating conditions. There are various ways in which the correlation between optimum L/D ratio and misalignment angle may be illustrated. The essential feature of the method chosen in Fig. 25 is that it facilitates clear graphical presentation of the results. In all the cases analysed, the load (W) was adjusted to give $h_{min} = 0.2$ mm at the bearing end with $L/D = 1.0$. The L/D ratio was then increased in steps, maintaining α and W constant. A family of curves for a range of α values was thus produced. Results are given for $C_d = 2$ and 1. mm., the former being consistent with normal sterntube bearing practice ($C_d/D = 0.002$). At $\alpha = 0$ and $C_d = 2$ mm, the above criterion ($h_{min} = 0.2$ mm at $L/D = 1.0$) was met by $W = 466\,000$ N. This was taken as the datum load, and all other loads are expressed as a percentage of this value.

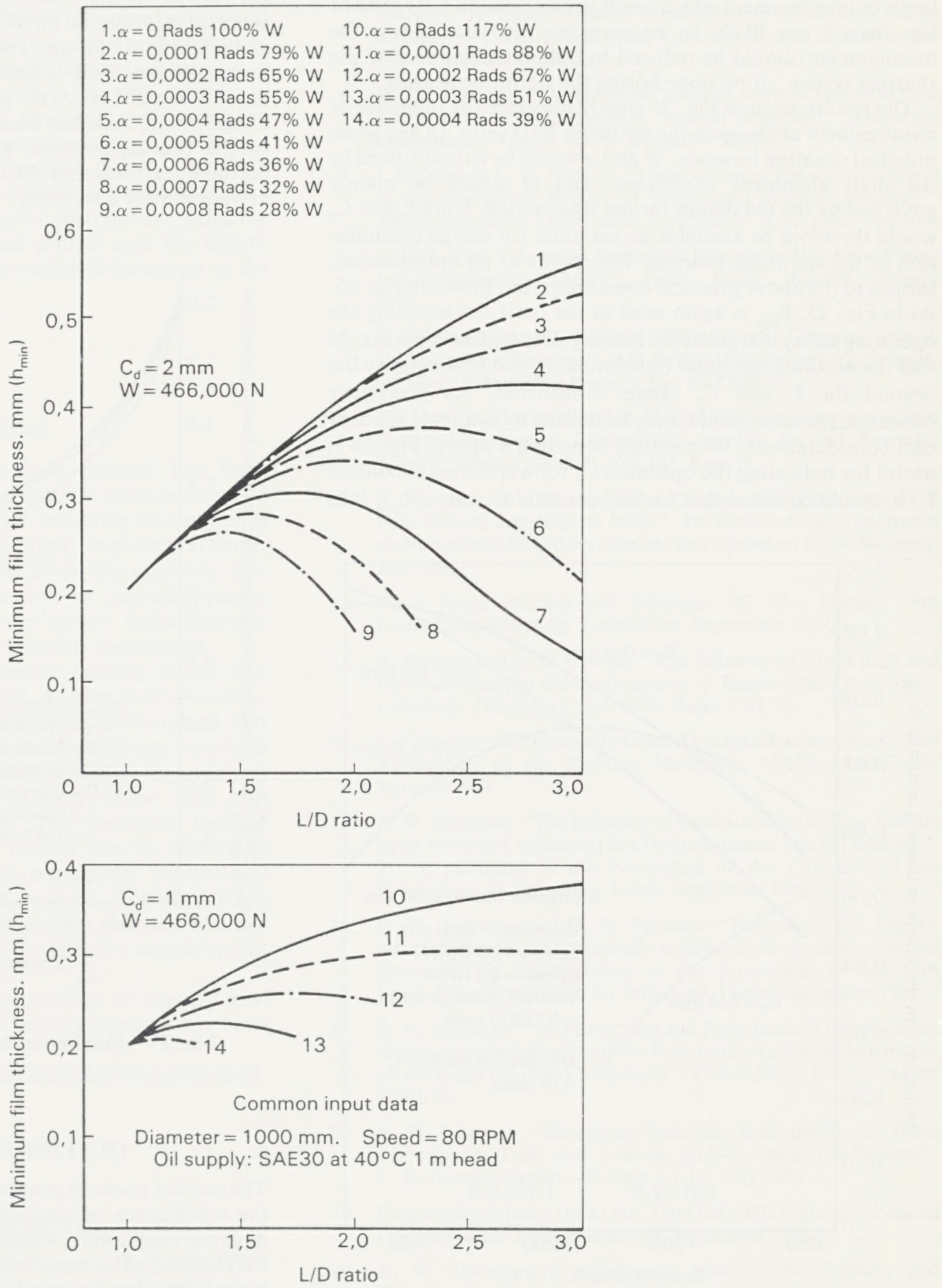


Fig. 25

For $C_d = 2$ mm, it is evident that α must exceed 0.0005 rads before there is any incentive to reduce L/D below 2.0. Reducing C_d to 1 mm increases the load capacity at $\alpha = 0$ due to the enhanced hydrodynamic effect. This improvement is substantially offset by the higher operating temperature, and hence reduced effective viscosity resulting from the tighter clearance. A more pronounced fall in Q, as L/D increased, was found for $C_d = 1.0$ mm, hence the much flatter family of curves due to the above thermal effects becoming even more significant. The direct effect of misalignment is more pronounced at reduced C_d, since it is essentially dependent on dimensionless misalignment $\overline{\alpha}\,(= \alpha\, L/C_d)$.

At $C_d = 1$ mm, the misalignment threshold beyond which it is worthwhile to reduce L/D below 2.0 is about 0.00015 rads. These results clearly indicate the need for a reasonably generous clearance in sterntube bearings, particularly where higher levels of misalignment are predicted. Given an adequate clearance, levels of misalignment which result in an optimum L/D ratio of less than 2 are likely to be excessive. In such cases, the misalignment should be reduced by suitable alignment of the shafting system, or by slope boring the sterntube bearing.

The results given in Fig. 25 clearly illustrate the influence of misalignment angle upon the optimum L/D ratio. In any given practical situation however, W and α would be virtually fixed by the shaft alignment conditions, and D would be mainly governed by the maximum torque transmitted. Only L and C_d would therefore be available as variables for design optimisation of the sterntube bearing. The results of an optimisation, subject to the above practical constraints, are shown in Fig. 26. As in Fig. 25, h_{min} is again used as the basis for assessing the operating safety margin of the bearing. It is evident from Fig. 26 that the absolute optimum (maximum h_{min}) in this instance lies beyond the L and C_d ranges considered. As previously indicated, the maximum L may be limited by available space or cost considerations. Where such constraints apply, Fig. 26 is useful for indicating the optimum C_d for a specified maximum L. It should be noted that for any constant C_d curve, h_{min} falls more rapidly if L is increased beyond the optimum value, compared with the fall below this point. If the misalignment angle is greater than predicted, the effect is to displace the operating point to the right of the optimum position on the appropriate constant C_d curve. It is therefore safer to select C_d a little above the optimum value in order to make some allowances for misalignment angle prediction error.

The results shown by Fig. 25 indicate an inverse correlation between the maximum acceptable specific bearing pressure and the maximum acceptable misalignment angle. In order to provide generalised guidance on this correlation, it is necessary to present the data in dimensionless terms. This has been done in Fig. 27 for a maximum eccentricity ratio at the bearing end of 0.9365. The choice of this eccentricity ratio is arbitrary, and should ultimately be related to service experience. A normal sterntube bearing C_d/D ratio of 0.002 was used to derive the curves in Fig. 27, but calculations indicated negligible differences for the range $0.0018 \leqslant C_d/D \leqslant 0.0022$. In the absence of specified information on oil type and operating temperature the assumption of an operating viscosity η of 0.033 Pa.s. is recommended. This is based on a SAE 30 oil at 60°C, and should generally err on the pessimistic side. It may be noted that a logarithmic scale has been used for dimensionless maximum specific bearing pressure. This form of presentation results in partial linearization of the curves, thereby enabling data to be extracted more accurately.

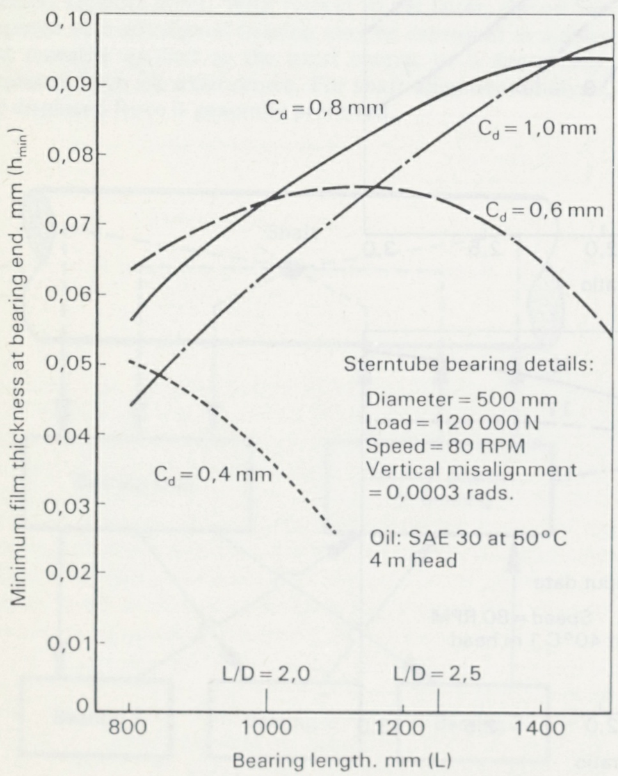


Fig. 26 Bearing Clearance and Length Optimisation

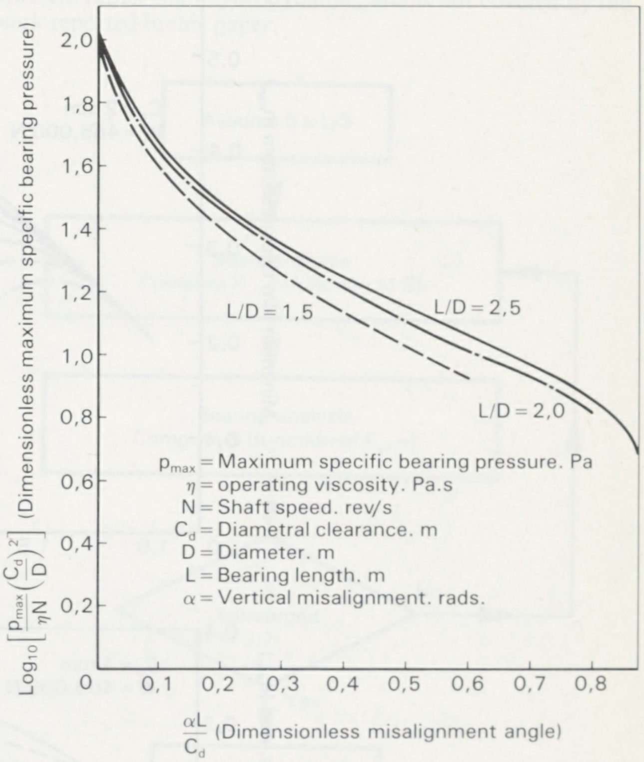


Fig. 27 Maximum Bearing Pressure for $\epsilon = 0{,}9365$

6. OUTLINE OF FUTURE WORK

The current research programme includes an investigation into the significance of various combinations of the 32 oil film dynamic coefficients, which may be computed by the program "STBSDC". This work will make comparisons of the predicted lateral vibration frequencies, when applying different combinations of coefficients to the analysis. The tests will be conducted on various types of propeller shafting system.

It is also desirable to investigate the non-linear response of sterntube bearings to lateral vibration, which is relevant to situations involving large amplitudes. This could be done by extending the journal orbit analysis method developed for aligned crankshaft bearings, to cover sterntube bearings under dynamic misalignment conditions. In the longer term, a substantially simpler approximate solution to this problem will be necessary, in order to reduce the computation time to a level acceptable for practical application.

Two other areas that are worth investigating are bearing elasticity (elastohydrodynamic lubrication) which is particularly relevent to reinforced resin bearings, and consideration of optimum oil groove locations. As indicated in section 2.6., the oil groove locations must be related to the load direction for a given installation and operating condition.

The research on sterntube bearings described in this paper, has, to date, been entirely theoretical. Comparisons of results with other published theoretical and experimental data have been carried out. The availability of clear reliable experimental data on misaligned bearings in general; and sterntube bearings in particular is, however, extremely limited. There is a consequent need for experimental work in this area, in order to improve confidence in the analysis programs that have been developed. Whilst shipboard measurements may give useful insights into the behaviour of sterntube bearings in their operating invironment, they are no substitute for good quality experimental test rig data. A bearing test rig offers the scope for comprehensive instrumentation, and control over the significant variables. It is hoped that such work will form part of the continuing research in this field.

7. CONCLUSIONS

A numerical hydrodynamic analysis method has been developed for journal bearings. This method takes account of flow continuity throughout the oil film, including the cavitation zone. The cavitation model is a particularly important feature, when considering the non-linearity of an oil film response. For the various aspects of the application of journal bearing analysis to practical situations, the above hydrodynamic analysis method has proved to be a valuable foundation.

The significance of various sterntube bearing design and operating parameters has been studied. As a result of this work, programs covering the steady load performance and oil film dynamic coefficients of misaligned sterntube bearings have been developed. These programs have been designed for practical applications using the Hewlett Packard 9836c desk top computer, and are user friendly. The sterntube bearing performance program is now in regular use to assess the acceptability of bearing operating conditions, particularly where high misalignment angles are involved. This program is also used in conjunction with shaft alignment analyses, in order to predict accurately the location of the effective support point in oil lubricated sterntube bearings.

Generalised guidance on the acceptability of angular misalignment as a function of specific bearing pressure has been given.

An outline of future work in this field has also been presented.

8. ACKNOWLEDGEMENTS

The author is indebted to Mr. S. G. Pratt for the provision of statistical data on sterntube bearing and aft seal defects, to Mr. G. Pumphrey for the preparation of the figures and to Mrs. T. Williams for typing the manuscript.

9 REFERENCES

1. C. Archer and D. K. Martyn: "Static and Dynamic Alignment" Trans.I.Mar.E. Vol.91, Conf. No. 4. Paper C31, (1979).
2. W. S. Vorus and R. L. Gray: "Calculation of Sterntube Bearing Eccentricity Distribution: Applications in States Lines Ro/Ro Bearing Failure Analysis". Trans. I.Mar.E. Vol.91, Conf. No. 4. paper C37, (1979).
3. M. Leemens and F. Roode: "Vibration Analysis of a Rotating Propeller Shaft in a Rubber Stern Tube Bearing". Conf. International Co-operation on Marine Eng. Systems. Trieste. (1984).
4. B. Fantino., J. Frene and M. Godet: "Dynamic Behaviour of an Elastic Connecting Rod Bearing—Theoretical Study". SAE/SP-539—Studies in Engine Bearings and Lubrication, Paper No. 83037. February 1983.
5. G. A. La Bouff and J. F. Booker: "Dynamically loaded journal bearings: A Finite Element treatment for rigid and elastic surfaces". ASME/ASLE Joint Tribology Conference. October 1984. Paper 84-TRIB-II.
6. R. W. Jakeman: "A numerical analysis method based on flow continuity for hydrodynamic journal bearings". Tribology International Vol.17. No. 6. December 1984.
7. I. Etsion and L. P. Ludwig: "Observation of Pressure Variation in the Cavitation Region of Submerged Journal Bearings." ASME Journal of Lubrication Technology. April 1982. Vol.104.
8. C. H. T. Pan: "Dynamic Analysis of Rupture in Thin Fluid Films. 1.—A Noninertial Theory". Trans. ASME. Vol.105. January 1983.
9. H. Heshmat and O. Pinkus: "Mixing Inlet Temperatures in Hydrodynamic Bearings". ASLE/ASME Tribology Conference. October 1985. ASME Preprint No. 85-TRIB-26.
10. T. Lloyd., R. Horsenell and H. McCallion: "An investigation into the Performance of Dynamically Loaded Journal Bearings: Theory". Proc.I.Mech.E. Vol.181. Part 33.
11. R. W. Jakeman: "Journal Orbit Analysis taking account of Oil Film History and Journal Mass". 4th International Conference on Numerical Methods in Laminar and Turbulent Flow. Swansea. July 1985.
12. G. J. Jones: "Crankshaft Bearings: Oil Film History" 9th Leeds-Lyon Tribology Symposium. September 1982.
13. R. Holmes and A. H. Craven: "The Influence of Crankshaft and Flywheel Mass on the Performance of Engine Main Bearings". I.Mech.E. Tribology Conf. 1971. Paper C63/71.
14. J. F. Booker: "Dynamically Loaded Journal Bearings: Numerical Application of the Mobility Method". ASME J.Lub.Tech. January 1971.
15. R. W. Jakeman: "The Influence of Cavitation on the Non-Linearity of Velocity Coefficients in a Hydrodynamic Journal Bearing". To be published in the Proceeding of the 13th Leeds-Lyon Symposium on Tribology. Leeds. September 1986.
16. R. W. Jakeman and D. W. Parkins: "Theoretical and Experimental Orbits of a Dynamically Loaded Hydrodynamic Journal Bearing". To be published in the proceeding of the 13th Leeds-Lyon Symposium on Tribology. Leeds. September 1986.
17. R. H. Bannister: "A Theoretical and Experimental Investigation Illustrating the Influence of the Non-Linearity and Misalignment on the Eight Oil Film Coefficients". Proc.I.Mech.E. 1976. paper C219/76.
18. R. W. Jakeman: "Misaligned Sterntube Bearings: Steady Load Performance Data and Stiffness and Damping Coefficients". L. R. Internal Report—Project 5.1.10. July 1984.
19. Engineering Science Data Unit. Item No. 66023. Steadily Loaded Hydrodynamic Journal Bearings. September 1966.
20. R. W. Jakeman: "Performance Analysis and Stiffness and Damping Coefficients for Misaligned Sterntube Bearings: User Guidance for Programmes STBPER and STBSDC". L. R. Internal Report. Project 5.1.10. June 1985.

APPENDIX 1

NOMENCLATURE

A_{xx} etc.	Oil film stiffness coefficients	See equations [1] [2] in section 3.5.
B_{xx} etc.	Oil film damping coefficients	
B_{tt} etc.	Oil film velocity coefficients	See equations [3] [4]
B_{ow}	Displacement of journal axis at axial centre from straight line joining axis locations at bearing ends.	
C_d	Diametral clearance.	
D	Journal diameter.	
e	Journal eccentricity†	
F	Total oil film force*	
F_r	Oil film radial force†	
F_t	Oil film tangential force†	
F_x	Oil film horizontal force*	
F_y	Oil film vertical force*	
H	Power loss.	
h_{min}	Minimum oil film thickness	
L	Bearing length	
M	Total oil film moment*	
M_x	Oil film moment in horizontal plane*	
M_y	Oil film moment in vertical plane*	
N	Angular velocity of journal about its axis. rev/s.	
p_h	Oil head pressure.	
Q	Oil flow rate from bearing ends.	
$\dot{R}$	Radial velocity of journal†	
u	Journal surface velocity.	
V_n	Normal velocity of journal surface at any given oil film element.	
W	Bearing load (external)	
x,y	Journal horizontal, vertical lateral displacement*	
$\dot{x}$,$\dot{y}$	Journal horizontal, vertical lateral velocity*	
α	Misalignment angle.	
α_x, α_y	Misalignment angle components in horizontal, vertical planes*	
β	Angle of misalignment plane*	
λ, γ	Journal angular displacement in horizontal, vertical plane*	
$\dot{\lambda}$, $\dot{\gamma}$	Journal angular velocity in horizontal, vertical plane*	
$\dot{\Theta}$	Angular velocity of journal axis about bearing axis†	
$\dot{\Theta}_o$	Equivalent angular velocity of journal†	
Θ_f	Angle of total oil film force*	
Θ_m	Angle of plane of total oil film moment*	
Δa, Δc	Axial, circumferential oil film element dimensions.	
ϵ	Eccentricity Ratio. ($=2e/C_d$).	
ψ	Attitude Angle*	
ω	Angular velocity of journal about its axis rad/s.†	
η	Effective viscosity.	

*see Fig. 2 †see Fig. 5.

Dimensionless parameters are denoted by a "bar" above them, and are defined as follows:

$$\overline{W} = \frac{W}{\eta NLD}\left(\frac{C_d}{D}\right)^2$$

$$\overline{M} = \frac{M}{\eta NL^2D}\left(\frac{C_d}{D}\right)^2$$

$$\overline{H} = \frac{HC_d}{\eta N^2LD^3}$$

$$\overline{Q} = \frac{Q\eta L^2}{WC_d^{\,3}}$$

$$\overline{p}_h = \frac{p_hLD}{W}$$

$$\overline{\alpha} = \frac{\alpha L}{C_d}$$

$$\overline{B}_{ow} = \frac{2.B_{ow}}{C_d}$$

$$\overline{A}_{xx} = \frac{A_{xx}C_d}{W}, \text{ etc.} \qquad \overline{B}_{xx} = \frac{B_{xx}\omega C_d}{WD}, \text{ etc.}$$

$$\overline{A}_{x\lambda} = \frac{A_{x\lambda}C_d}{WL}, \text{ etc.} \qquad \overline{B}_{x\lambda} = \frac{B_{x\lambda}\omega C_d}{WLD}, \text{ etc.}$$

$$\overline{A}_{\lambda x} = \frac{A_{\lambda x}C_d}{WL}, \text{ etc.} \qquad \overline{B}_{\lambda x} = \frac{B_{\lambda x}\omega C_d}{WLD}, \text{ etc.}$$

$$\overline{A}_{\lambda\lambda} = \frac{A_{\lambda\lambda}C_d}{WL^2}, \text{ etc.} \qquad \overline{B}_{\lambda\lambda} = \frac{B_{\lambda\lambda}\omega C_d}{WL^2D}, \text{ etc.}$$

APPENDIX 2

GLOSSARY

Adhered cavitation model — Cavitation model in which oil is assumed to be transported through the cavitation zone in a layer adhered to the moving (usually journal) surface. (*See* Fig. 3.)

Boundary lubrication — Lubrication regime in which load is carried partly by surface contact and partly by hydrodynamic action.

Cavitation pressure — Minimum film pressure, i.e. that pressure at which gas or vapour bubbles start to form thus preventing any further fall in pressure.

Cross axis coupling — Refers to hydrodynamic bearing characteristic that journal displacement or velocity in one direction induces oil film force components in direction at right angles.

Damping coefficient — Linearised rate of change of oil film force per unit change in journal velocity. Also referred to as velocity coefficient.

Dynamic coefficient — Collective term for stiffness and damping coefficients.

Dynamic misalignment — Angular misalignment of journal relative to bearing subject to cyclic variation resulting from dynamic load.

Elastohydrodynamic lubrication — Hydrodynamic lubrication in which elastic deformation of journal and/or bearing surfaces due to oil film pressure is significant.

Element grading — Variation of oil film element dimensions to give better modelling and thus improved accuracy of numerical oil film pressure solution.

Gaseous cavitation — Cavitation due to dissolved gas in oil coming out of solution.

Gauss Seidel relaxation method — Numerical solution method used to determine oil film pressure distribution. Oil film element pressures are successively computed in terms of current element pressures for adjacent elements until convergence is achieved.

Hydrodynamic lubrication — Lubrication in which journal and bearing surfaces are completely separated by lubricant film. Load is carried by hydrodynamic pressure generated in the lubricant film.

Journal — That part of the shaft within the bearing.

Journal orbit — Displacement path traced by journal centre in a dynamically loaded bearing.

Minimum film thickness — Minimum separation of journal and bearing surfaces. Used as a criterion to assess the acceptability of bearing operating conditions.

Misalignment — Angle between journal and bearing axes. May be expressed as total value or vertical and horizontal components. Where the journal axis is bowed, the mean angle over the length of the bearing is taken.

Mobility Method — Method devised by J. F. Booker for carrying out a fast journal orbit analysis. (*See* reference 14.)

Oil film element — Small section of oil film of rectangular planform. Oil film is divided into such elements for numerical solution of the pressure distribution.

Oil film history — Concept in the modelling of the oil film in a dynamically loaded bearing. This takes account of the fact that cavitation zones take a finite time to disappear after the journal displacement and velocity conditions causing them are changed.

Over relaxation — Technique for obtaining fast convergence in the Gauss-Seidel relaxation process.

Perturbation — Small displacement or velocity increment applied to the journal for the purpose of computing linearised stiffness and damping coefficients.

Reaction Method — Fast journal orbit analysis method devised by the author. (*See* references 15 and 16.)

Reformation boundary — Downstream boundary of cavitation zone at which full oil film reforms.

Rupture boundary — Upstream boundary of cavitation zone at which full oil film ruptures.

Short bearing approximation — Approximate analytical solution of Reynold's equation which assumes that circumferential film pressure gradients are negligible relative to axial pressure gradients.

Specific bearing pressure — Mean bearing pressure based on load divided by projected area, i.e. W/LD.

Sterntube bearing — Bearing located within sterntube supporting propeller shaft. In relation to loading conditions and the work covered by this paper, it refers to the bearing adjacent to the propeller and may therefore include "A" bracket bearings, etc.

Stiffness coefficient — Linearised rate of change of oil film force per unit change in journal displacement. Also referred to as displacement coefficient.

Striated cavitation model — Cavitation model which assumes that oil is transported through the cavitation zone in rectangular section streams extending from the journal to bearing surface. (*See* Fig. 3.)

Squeeze action — Generation of hydrodynamic film pressure by the component of lateral journal velocity in the direction of the line connecting the bearing and journal centres.

Vapourous cavitation — Cavitation resulting from the formation of vapour bubbles.

Ventilation — Form of gaseous cavitation where gas (usually air) is drawn from outside the bearing oil film. This clearly cannot occur in a fully submerged bearing.

Viscosity — In the context of this paper this refers to dynamic viscosity, which may be defined as the lubricant shear stress per unit velocity gradient. This is assumed to be constant at a given temperature and pressure, i.e. Newtonian lubricant. This is an approximation since in reality viscosity varies with the magnitude of the velocity gradient, an effect referred to as shear thinning. In normal journal bearing conditions the effects of shear thinning and pressure upon viscosity are negligible.

Wake field — Distribution of water velocity around the aft end of a hull. This interacts with the propeller to produce thrust and torque eccentricity.

Wedge action — Generation of hydrodynamic film pressure, due to journal surface velocity inducing lubricant into the convergent space between the journal and bearing surfaces.

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Discussion

on the paper

STERNTUBE BEARINGS: PERFORMANCE CHARACTERISTICS AND INFLUENCE UPON SHAFTING BEHAVIOUR

by

R. W. Jakeman

Paper No. 2. Session 1986–87

Any opinions expressed and statements made in this Discussion Paper are those of the individuals.

Hon. Sec. R. V. Pomeroy
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Discussion on the Paper

STERNTUBE BEARINGS:

PERFORMANCE CHARACTERISTICS AND INFLUENCE UPON SHAFTING BEHAVIOUR

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R. W. Jakeman

DISCUSSION

From Mr. P. F. C. Horne:

At the meeting I said that the original Rule of L/D = 4 was determined by service experience nearly 100 years ago. We should not lose sight of service experience today. Some figures extracted by TRO for a submission to IACS by Mr. Siggers are worth considering. These have been graphed as Figs. D1–D4. It will be seen that there is no evidence to support going to very short bearings, even if the Reinforced Resin bearing materials are excluded.

The Author showed the effects of varying each of several parameters independently and showed the advantage of short bearings. This should, however, be considered in the light of the limits of screwshaft diameter. Except where TVC's dictate, it seems unlikely that a screwshaft significantly in excess of Rule diameter would be acceptable to an owner. Thus reduction in L/D implies an increase in bearing pressure in almost all practical applications. It would be interesting to see what Fig. 12 in the paper would look like for a constant shaft diameter.

It seems possible that an increase in bearing loading might well have greater adverse effect under boundary lubrication conditions and could also defer the onset of a hydrodynamic lubrication regime. I wonder whether any consideration has recently been given to the use of hydrostatic lubrication for conditions where a hydrodynamic film has not been established. I believe some work was done on such systems a few years ago and wonder whether any advances have been made.

Large diameter shafts such as have been fitted to some modern high powered low speed systems must be much more rigid in comparison with the stiffness of the bearing supports than with smaller diameter shafts. The alignment calculations as in Ref. 1 must be of doubtful value in such cases. Has the author any information on alternative calculation methods?

From Dr. M. A. Kavanagh:

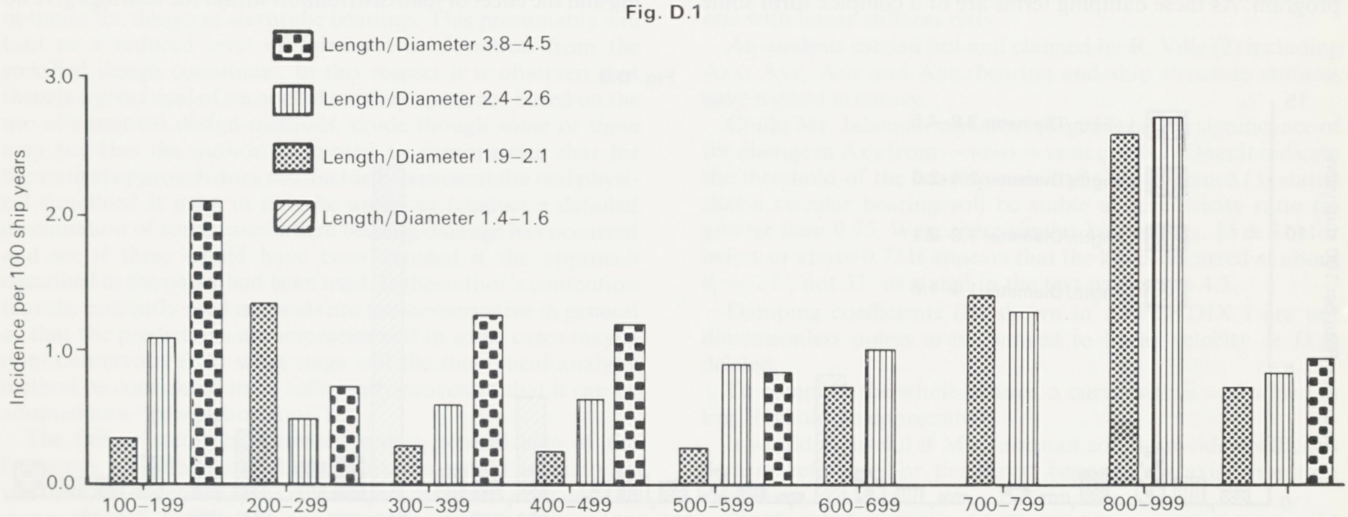
Answer to the question by Mr. Kunz concerning the likely ill conditioning problems when introducing the stiffness and damping terms, obtained from Mr. Jakeman's oil film program, into a vibration analysis of a complete shafting system:

Preliminary work has been carried out to investigate the effects of introducing the additional stiffness coefficients, as derived from the oil film program, into a NASTRAN finite element model of a complete shafting system. The effect of these terms on the resultant value of the critical vibration frequencies was recorded for a lateral vibration analysis.

At the moment when carrying out a lateral vibration analysis a lateral spring stiffness is introduced at the stern bearing. In this investigation two additional spring coefficients were applied, a rotational spring and a cross coupling spring, that is a spring that produces a force due to rotation and a couple due to a displacement.

The gyroscopic effects have been switched off as these additional terms will only affect the overall location of the criticals.

Fig. D.1



TRO Investigation 1892
Tailshaft Sternbush Bearings
Oil Lubricated Bearings
Other than Synthetic Resin

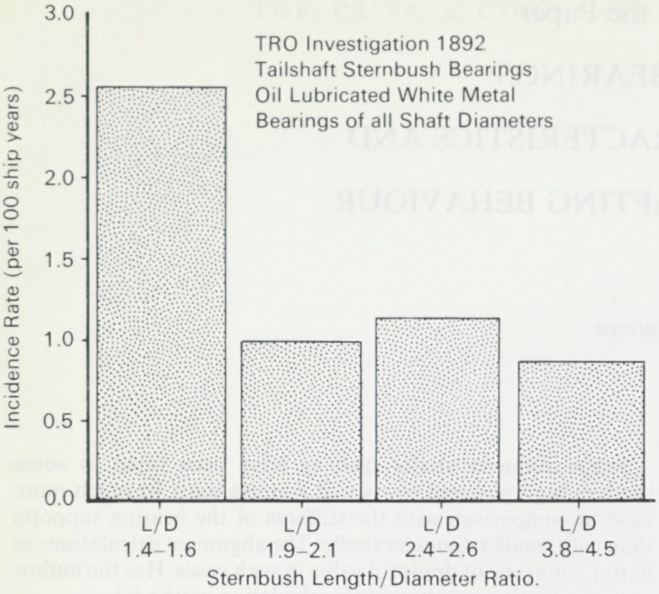


Fig. D.2

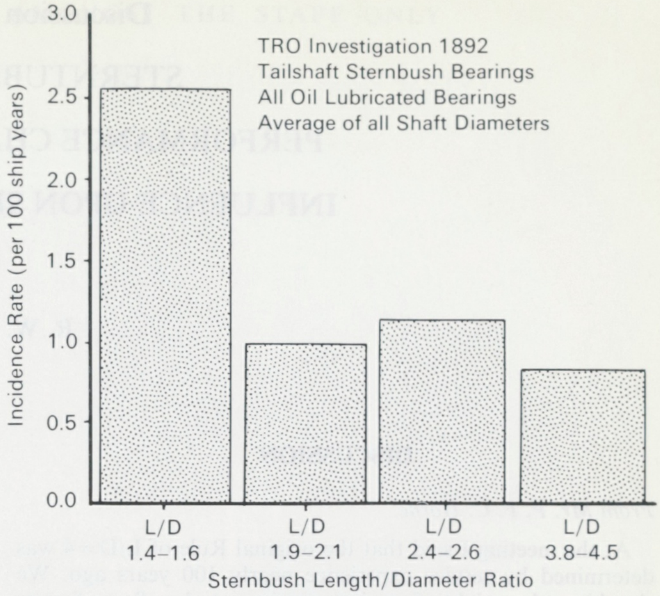


Fig. D.4

Below are listed a summary of the effects of these stiffness terms on the first critical frequency on a typical shaft system:

1. Rigid stern bearing $W_1 = 362$ CPM
2. Lateral spring at the stern bearing $W_1 = 320$ CPM
3. Lateral spring and rotational spring at the stern bearing $W_1 = 353$ CPM
4. Lateral spring and rotational spring and coupling spring $W_1 = 325$ CPM

The results of cases 1 to 3 produce a predictable change, with the first critical increasing as the effective stiffness increases. When the cross coupling term is included however, Case 4, the effective stiffness of the system has been decreased. The sign of the cross coupling term, which will determine its contribution to the effective stiffness, is dependent upon the overall configuration of the sterntube bearing. Consequently the overall effect of all these additional stiffness terms is uncertain and can only be fully assessed in this manner for each specific shafting system.

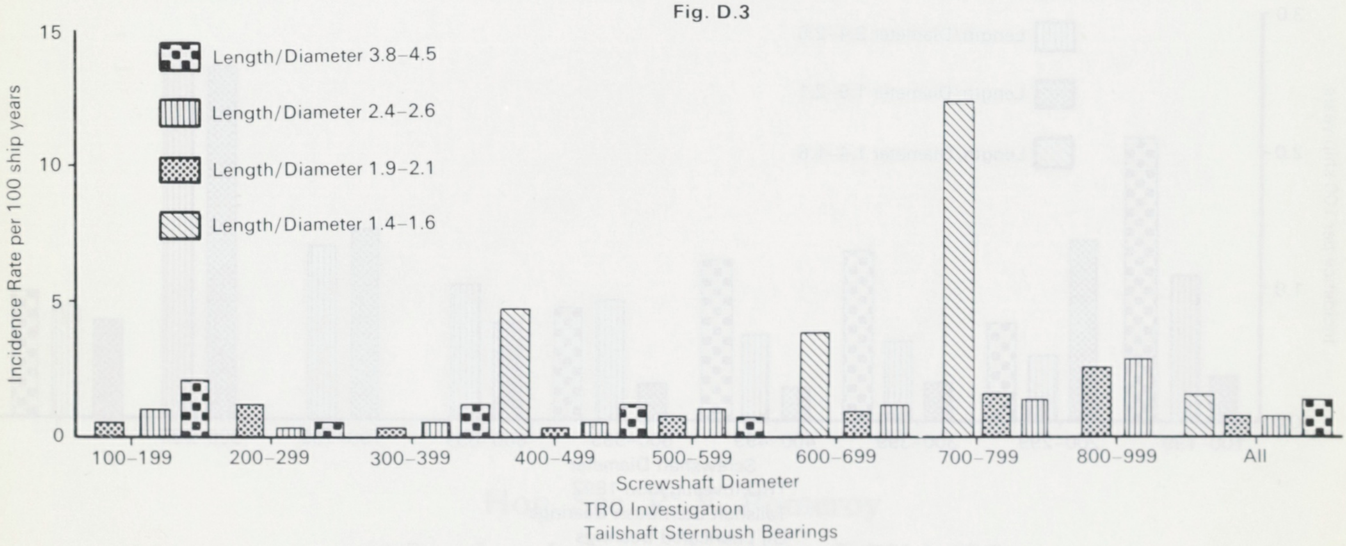
With regards to ill conditioning of the solutions matrices, no problems were envisaged or experienced with the introduction of these stiffness terms. The next stage in this work however is to introduce damping terms which are also provided by the oil film program. As these damping terms are of a complex form some problems may be experienced. In addition these terms may also cause problems when the gyroscopic effects are switched on since they are also of a complex form.

From Mr. F. Kunz:

Mr. Jakeman is to be thanked for an interesting paper on an important topic and for the considerable effort which must have gone into the preparation of it. It is noted that another area where the Society has expended a sizeable theoretical effort over a number of years has yielded results which are in a form ready for assessment in practical applications.

The 32 coefficients in equation 2 are somewhat overwhelming and I note that more work is proposed to assign numerical values to some of them. Solutions of matrix equations always raise questions of sensitivity and maybe Mr. Jakeman could put my mind at rest by commenting on the likely effects of small changes in input values on the solution. It is noted that the application of the coefficients to shaft lateral vibrations remains to be explored. This would be a worthwhile task because current natural frequency calculations which ignore excitation, damping and the effect of load distribution within the bearings give no

Fig. D.3



help in assessing the significance of any calculated criticals. Experimental evidence of the importance of the sterntube bearing geometry or material on the vibration levels at the sterntube outboard seal has been produced by the Technical Investigations Department. For example, a simple change of bearing material from a resin type bearing to a conventional white metal bearing of the same dimensions but reduced clearance reduced vibrations at the outboard seal by a factor of about two while the sterntube oil leakage which promoted the investigation was reduced from about two hundred litres of oil per day to negligible amounts. This solution to a very significant problem had to be arrived at on a mainly intuitive basis and the use of a rational basis would have saved much concern and help in future cases.

Experience of this type makes me share Mr. Jakeman's views on resin impregnated bearings but it remains a fact that a large number of such bearings perform satisfactorily particularly on larger vessels. It would have been useful to include ship's size in the variables explored for correlation with failure incidence but maybe Mr. Jakeman has already done this. I find the statement that sterntube seal defects invariably result in damage to the bearing rather sweeping as many seals receive attention because of leakage which in the nature of things can be of sea water into the bearing or loss of oil to the sea.

Figure 27, to my mind, is one of the most significant results of Mr. Jakeman's work as it models the effects of internal bearing misalignment, clearance and bearing length. I take it to indicate that length is a somewhat secondary factor, a conclusion which probably should be tempered with the concept that longer bearings may reduce lateral vibration response if well aligned. The great merit of FIGURE 27 is, however, that it shows relationships which can be put to the test and could be modified in the light of experience.

In section 5.2 the statement that misalignment is generally insignificant is at first sight rather challenging and at variance with experience. Maybe it is intended to apply to angular deviations of the centre lines of bearings and shafts rather than the deviations of bearings from their intended relative lateral positions normally associated with misalignment. While it is true that short bearings are less sensitive angular errors between bearing and shaft axes quite a number of problems are known to have arisen from this cause and bedding checks remain valid.

From Mr. R. V. Pomeroy:

The author has shown in this paper how a collection of useful characteristic parameters can be derived theoretically. There is a clear implication that the results of the analysis can be used to optimise the design of sterntube bearings. This presumably will lead to a reduced level of tolerance to deviation from the specified design conditions. In this respect it is observed that there is a great deal of successful service experience based on the use of empirical design methods, crude though some of these may be. Has the author attempted to demonstrate that his theoretical approach does satisfactorily represent the real physical situation? It may, in fact, be useful to conduct a detailed examination of some cases where bearing damage has occurred and see if these would have been avoided if the approach described in the paper had been used. Is the author's contention that the presently used methods are too conservative in general or that the predictions are imprecise and in some cases maybe non- conservative? At what stage will the theoretical analysis method be considered to be sufficiently proven so that it can be adopted as a "production tool"?

The failure statistics presented give a general idea of the frequency of defects but further analysis would seem to be worthwhile to identify areas where most problems arise and the causes thereof. Firstly it is not clear why shaft diameters less than 400mm have been excluded. Does the failure rate continue to decrease as shaft diameter reduces for oil-lubricated bearings? Secondly, what is meant by defect? It would be very useful to know how many of the reported defects were minor and repairable and how many required renewal of the bearing. Perhaps even more interesting would be the number of cases where bearing failure directly results in further damage, to the tailshaft for instance. Although the author appears concerned by the defect rate it is noted that sterntube bearing failures did not feature in Mr. Munro's paper to the LRTA and only one recent case, a resin bearing, is reported in N.D.L. Without an analysis of cause and effect it is not reasonable to draw any conclusions from the information in the paper. Analysis of the data could serve to indicate which are the most useful areas to concentrate on in the continuing research in this subject.

In the paper there is no indication as to the affect of the machinery installation type on the performance of sterntube bearings. Is there any evidence to suggest that the service history is significantly different for:

i) steam turbine ships as opposed to diesel ships,
ii) geared installations as opposed to direct drive,
iii) controllable pitch as opposed to fixed pitch propellers,
iv) multiple as opposed to single screw.

The author has clearly devoted considerable effort in this subject area. It would appear that a reasonably robust calculation method has been developed. If the dynamic results provide a better understanding of shafting vibration then a clear advance will have been made. If however the only end result is that the old-fashioned empirical basis for design is about right all will not be in vain – industry expects that simple design rules can be substantiated! In fairness, with a mature product it would be surprising if any radical design changes are the result of this type of analysis. What undoubtedly improves is the fundamental understanding of the problem and this has been amply demonstrated by the author in this parametric study.

From Mr. W. Y. Ng:

I would like to offer my congratulations to Mr. Jakeman on the presentation of this very informative and comprehensive paper which will be a valuable addition to the study of oil lubricated sterntube bearings.

It is now the time to apply these data to lateral vibration analysis which due to the complexity of the shafting support and also for economical reasons, is possibly one of the least researched areas.

As a matter of interest to find out the influence of constraints on the natural frequency other than the single simple support treatment, Jasper's method (1) was used for a two-support shaft, the calculated natural frequency with linear stiffness (Ayy) together with angular stiffnes (Arr) was over 4% higher than the one with linear stiffness only.

An analysis carried out and claimed by R. Ville (2) including Axx, Ayy, Axy and Ayx, bearing and ship structure stiffness gave a good accuracy.

Could Mr. Jakeman comment (Fig. 17) on the significance of the change in Axy from $-$ve to $+$ve at $\varepsilon = 0.7$. Does it indicate the threshold of the bearing stability? One reference (3) stated that a circular bearing will be stable at eccentricity ratio (ε) greater than 0.75. Were there similar kinks (Figs. 15 & 21) for below or above 0.7? It appears that the kinks occurred at about $\theta_f = 27^\circ$, not 37° as stated in the text paragraph 4.3.

Damping coefficients (B) shown in APPENDIX I are not dimensionless unless ω is changed to linear velocity or D is deleted.

To complete the whole picture, a curve with $\bar{\alpha} = 0$ added in Fig. 9 would be appreciated.

It would be useful if Mr. Jakeman could provide coefficient data or reference for the thrust bearings for axial vibration analysis.

The Society has accepted L/D = 2 for some water lubricated synthetic bearings based on the hydrodynamic lubrication principle, it would be interesting to know if a bearing having this ratio has been installed in any ship. Presently all the latest Canadian Icebreakers have bearings with L/D $\geqslant$ 4. It is quite

possible that water lubricated bearings which have the advantage of simplicity and no risk of pollution on failures, will become popular once more.

Oil film stiffness data given in this paper are in good agreement with those in R. Ville's paper.

Reference

1) Norman H. Jasper L.S.p Vol 3 No.20-1956
2) R. Ville ICMES '84 Conference in Trieste
3) Handbook of Turbo Machinery (?) A21

AUTHOR'S REPLY

To Mr. Horne:

I agree that service experience should never be ignored since theoretical analyses are always an approximation to reality. The main value of theoretical work lies in the provision of a rational basis for assessing the relative influence of all the significant variables. It is essential to remember that theoretical predictions are by no means exact, and that the parameters predicted do not in themselves tell us precisely where failure will occur.

It is gratifying to note that Mr. Horne's records support my general conclusion that there is no incentive, with respect to safe operation, in going to shorter bearings. The data for all shaft diameters in Figs. D2 and D4 do, however, appear to disagree with the corresponding results in Figs. D1 and D3.

Mr. Horne's statement that I have shown the advantage of shorter bearings is only true for the very restricted situation of misalignment angles substantially in excess of the levels normally considered to be acceptable (see Fig. 25). Regarding the reference to Fig. 12, since this figure is entirely in dimensionless terms it is valid for any shaft diameter, constant or otherwise.

I have no knowledge of any recent developments with respect to the application of hydrostatic bearings to the sterntube bearing situation. I agree that higher bearing pressures are likely to have an adverse effect on low speed operation under boundary lubrication conditions, and to raise the speed at which hydrodynamic lubrication is attained. Although hydrostatic bearings enable full oil film lubrication to be achieved at low shaft speeds, the required pressurised oil supply poses certain problems in this application. If any sterntube bearing installation becomes dependent upon hydrostatic action, rather than simply improved by it, then a "fail safe" pressurised oil supply would be required.

Shaft alignment calculations taking account of bearing support flexibility can be carried out with the LR291 shaft alignment program. The flexible support facility has been little used due to the dearth of flexibility data. It should be noted that bearing oil films contribute towards bearing support flexibility, and this contribution can be estimated as shown in the paper. In addition, a steady state analysis version has been developed from my forced damped lateral vibration program, to which more detailed reference is made in the reply to Mr. Kunz. This enables a simultaneous vertical and horizontal alignment analysis to be carried out, taking account of all the bearing oil film stiffness terms shown in equation [2] of the paper. In comparison with previous shaft alignment analyses, this new program facilitates the inclusion of angular stiffness terms and vertical– horizontal cross coupling terms. Bearing support structural stiffness values can be combined with the corresponding oil film terms.

To Dr. Kavanagh:

I must thank Dr. Kavanagh for his interesting comments on the application of the oil film stiffness terms to lateral vibration analysis. My only addition to this is the cautionary note that oil films are inherently non-linear. When applying linearised oil film stiffness and damping coefficients there will be some degree of approximation depending upon the vibration amplitudes.

In addition, it should be noted that the various oil film stiffness and damping coefficients are determined by applying appropriate journal displacement or velocity perturbations one at a time, and computing the corresponding changes in the oil film force and moment components. The application of these coefficients to lateral vibration problems implicitly assumes that the principle of superposition applies to any combination of displacement and velocity components occurring simultaneously. The work reported in reference (15) indicated that the influence of cavitation in hydrodynamic bearings in fact negated the principle of superposition in this context. This results in a further source of error when using linearised dynamic coefficients.

To Mr. Kunz:

Mr. Kunz's question regarding the significance of the 32 oil film stiffness and damping coefficients has been partly answered by Dr. Kavanagh's comments, and there is nothing that I can add to these at present. Regarding the long standing need for a lateral vibration response prediction facility, I have in fact developed such a program since completion of the paper. This program models the shaft as a multi-mass-elastic system, incorporates alternative linear and non-linear bearing oil film models and is based on the time-stepping approach. Other features of this program are:

a) The complete propeller damping and entrained mass/inertia matrices are used.
b) Any form of propeller excitation may be specified, i.e. non-sinusoidal components of force and moment.
c) Elastic deflections of each shaft element include both bending and shear components.
d) Gyroscopic effects are considered on all elements.
e) Weight and buoyancy forces are also included.

Satisfactory operation of this program has been achieved. In its present form the program has been tailored to a particular test case corresponding to additional reference (D1). Further development work would be required to refine it to a form suitable for general application, and to improve computing time. There is no immediate prospect of the above development work being carried out since the research project, of which this work formed a part, has been terminated.

The TID case of the effect of changing from a resin to white metal sterntube bearing of reduced clearance was noted with interest.

Ship size was not specifically considered in the failure statistics, but is indirectly covered, albeit rather crudely, by the correlation with shaft diameter.

My statement that seal defects invariably result in bearing damage was, perhaps, just a little sweeping, as noted by Mr. Kunz. Where seal defects are relatively small and a positive oil head pressure is maintained, the bearing may escape damage. The lumping together of seal and bearing defects in much of the statistical data did, however, reflect the fact that these defects do frequently occur together.

Mr. Kunz's remarks on Fig. 27 are much appreciated. The use of a log function for dimensionless maximum specific bearing pressure does in fact make the influence of L/D ratio appear to be rather less than it really is. I agree with Mr. Kunz's view that Fig. 27 should be related to service experience. As noted in the paper, the maximum eccentricity ratio of .9365, used as the basis for computing the curves of Fig. 27, was somewhat arbitrary. The essential value of Fig. 27, lies in its format, in providing a clear guide to the relative significance of all the parameters, rather than the absolute magnitude of safe specific bearing pressures that may be derived from it.

I regret the confusion experienced by Mr. Kunz over the use of the term misalignment in section 5.2. Since hydrodynamic bearings will not support load without some lateral misalignment of the journal and bearing axes, it is reasonable to assume

that this type of misalignment will generally be present. In view of the fact that lateral misalignment may therefore be taken for granted, only angular misalignment has any particular significance, and is the only type of misalignment worthy of any specific reference. My usage of the term misalignment in this way is defined in the glossary in Appendix 2.

With regard to shorter bearings, although these are less sensitive to misalignment, they will still be subject to some limit for maximum acceptable misalignment. I would therefore endorse Mr. Kunz's view that alignment checks should be carried out for such bearings. Section 5.2 was, however, written more from the viewpoint of one performing an alignment analysis, where it is reasonable to assume that misalignment will be negligible in shorter bearings. This assumption is based not only on the reduced sensitivity to misalignment, but also on the fact that correction of misalignment in these bearings is generally fairly simple to carry out.

To Mr. Pomeroy:

Mr. Pomeroy mentions the possibility of reduced tolerance to deviation from specified design conditions, if a sterntube bearing design is optimised. Such a reduction would undoubtedly occur if one chose to exploit the optimisation by increasing the permitted load rather than accepting an increased safety margin. This problem also depends on the sensitivity of the bearing load capacity to the various parameters subject to optimisation. For example, the results given in Fig. 15 could be used to optimise the angular location of the oil supply grooves relative to the load vector. This figure shows a high degree of sensitivity around the optimum condition, with a particularly sharp fall in load capacity to the right of the optimum point as θ_f approaches 27°. It is generally assumed that the sterntube bearing load acts vertically downward ($\theta_f = 0°$), which corresponds to operation substantially to the left of the optimum point in Fig. 15. In view of the generally uncertain influence of the propeller–wake interaction on the load vector angle, the above non-optimum condition is preferable in maintaining a reasonable safety margin from the load capacity fall at $\theta_f = 27°$. The influence of the propeller–wake interaction in particular, renders the concept of specified design conditions in a sterntube bearing potentially dangerous (see reference (2)). When assessing the performance of any sterntube bearing, one must therefore make allowance for a fair degree of uncertainty in the actual operating conditions.

Regarding the question of whether the theoretical model satisfactorily represents the real physical situation, as noted in section 6 of the paper, useful measured data for sterntube bearings is somewhat sparse in the published literature. As mentioned in the reply to Mr. Kunz, measured data presented in reference (D1) has been used for correlation with the predictions of my forced-damped lateral vibration response program.

This data is unsuitable for correlation with steady load performance predictions due to the extent of dynamic load present and limited instrumentation. As a result of the above situation, my sterntube bearing performance predictions have only been correlated with those of other theoretical methods. These correlations have been reported in references (6) and (D2). Satisfactory correlation of predicted and measured data for crankshaft type bearings was reported in reference (16). This verified the validity of the hydrodynamic analysis method, the only significant parameter not covered being misalignment.

I would agree with Mr. Pomeroy's suggestion that it would be potentially instructive to carry out analyses of cases where bearing damage has occurred. The results would, however, be masked by the probability that in many such cases the bearing load and misaligment will not be known with any precision. In view of this problem a statistical approach would be appropriate. This would need to involve a large number of cases, both successes and failures, using the best available estimates for bearing load and misalignment in each case.

It would be fair to say that the present methods are a trifle imprecise in that they take account of but three parameters: specific bearing pressure, L/D ratio and (unofficially) misalignment angle. These are considered in isolation from each other by the simple specification of maximum or minimum permitted values. Fig. 27 of the paper shows how the seven relevant parameters may be applied in a rational manner.

The stage at which my theoretical method was adopted as a production tool was passed over two and a half years ago (see Section 7). To put this work into the correct perspective, it would be inappropriate to think in terms of the "Jakeman Theory" which needs to be proved. My work is essentially a refinement of well established theoretical concepts in this field, with specific adaption to the sterntube bearing situation. Within the limitations of the approximations made, this theory undoubtedly provides an adequate description of the way in which the various design and operating parameters interact to determine the bearing performance. The need to correlate performance predictions with service experience has already been covered in the reply to Mr. Horne.

In answer to Mr. Pomeroy's queries on the failure statistics, the 400mm shaft diameter cut off was arbitrary and determined by the availability of previously analysed data. It is noted that Mr. Horne's contribution provided data down to shaft diameters of 100mm. A "defect" is defined as any case reported by a surveyor as requiring remedial action. The extent of details reported is generally inadequate for the purposes of providing a severity breakdown of the statistics. It should be added, however, that defect severity is not necessarily related to importance, since today's minor defect may be tomorrow's total failure if not rectified. Of the possible factors influencing the failure statistics, as queried by Mr. Pomeroy, only the difference for controllable and fixed pitch propellers was investigated. As noted at the foot of Table 1, the type of propeller showed no significant correlation with defect rate. A general problem in this area was that breakdown of the data into groups, such as ship type, frequently reduced the numbers to a level that was statistically insignificant.

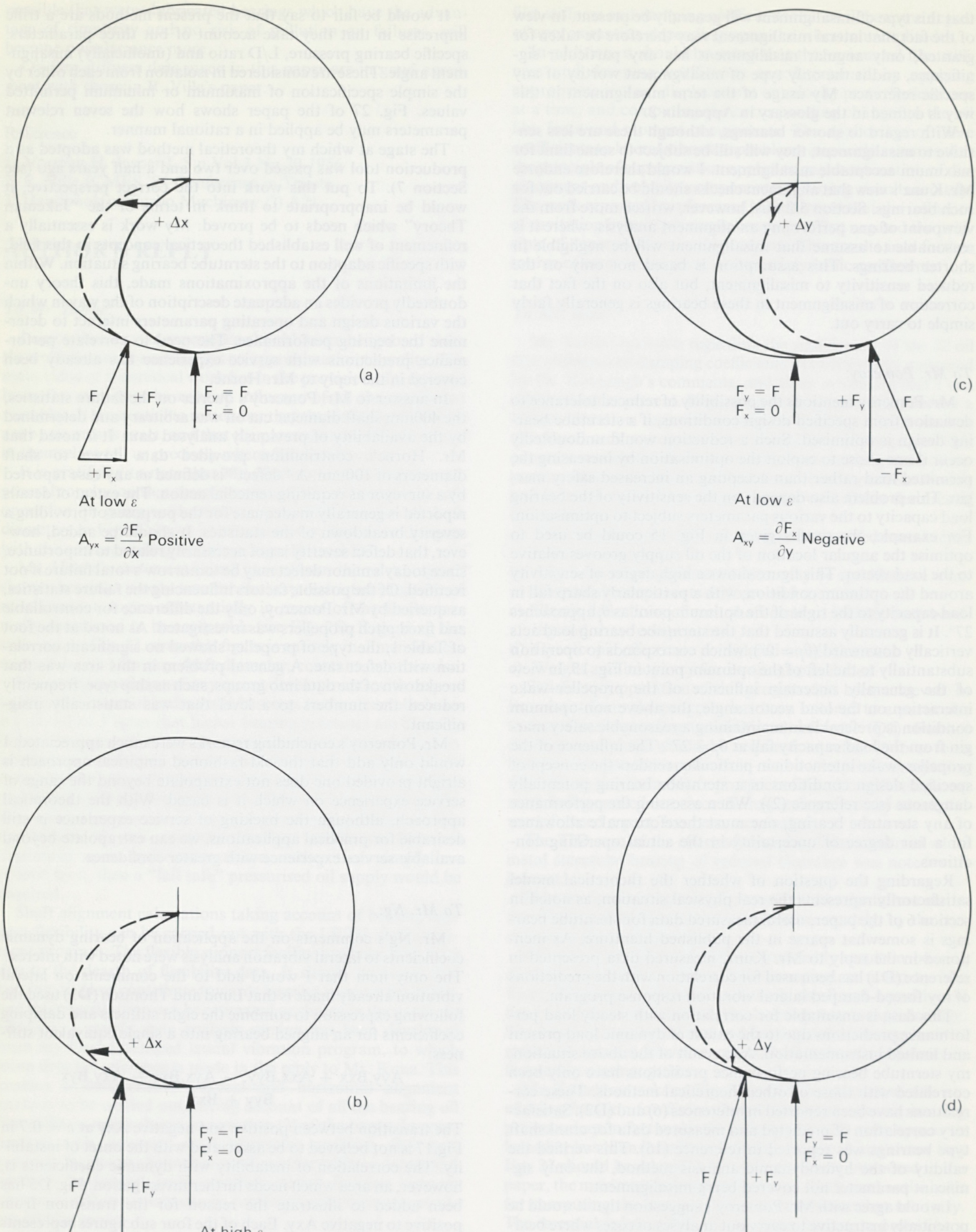
Mr. Pomeroy's concluding remarks were much appreciated. I would only add that the old fashioned empirical approach is alright provided one does not extrapolate beyond the range of service experience on which it is based. With the theoretical approach, although the backing of service experience is still desirable for practical applications, we can extrapolate beyond available service experience with greater confidence.

To Mr. Ng:

Mr. Ng's comments on the application of bearing dynamic coefficients to lateral vibration analysis were noted with interest. The only item that I would add to the comments on lateral vibration already made is that Lund and Thomsen (D3) used the following expression to combine the eight stiffness and damping coefficients for an aligned bearing into a single equivalent stiffness:

$$\frac{Ayy.Bxx + Axx.Byy. - Ayx.Bxy - Axy.Byx}{Byy + Bxx}$$

The transition between positive and negative Axy at $\varepsilon = 0.7$ in Fig.17 is not believed to be associated with the onset of instability. The correlation of instability with dynamic coefficients is, however, an area which needs further investigation. Fig. D5 has been added to illustrate the reason for the transition from positive to negative Axy. Each of the four sub figures represents the bearing clearance circle, i.e. the envelope within which the journal centre-line may move. Within the clearance circles, the solid crescent shaped curve represents the path traced out by the journal centre line when subject to a steady downward vertical applied load varying in magnitude from zero to infinity. This is referred to as the static journal locus. The forces shown are the corresponding equal and opposite oil film forces. When the



$+\Delta x$
(a)
$F_y = F$
$F_x = 0$
F
$+F_y$
$+F_x$
At low ε
$A_{yx} = \frac{\partial F_y}{\partial x}$ Positive
$+\Delta y$
(c)
$F_y = F$
$F_x = 0$
$+F_y$
F
F_x
At low ε
$A_{xy} = \frac{\partial F_x}{\partial y}$ Negative
$+\Delta x$
(b)
$F_y = F$
$F_x = 0$
F
$+F_y$
At high ε
$A_{yx} = \frac{\partial F_y}{\partial x}$ Positive
$+F_x$
$+\Delta y$
(d)
$F_y = F$
$F_x = 0$
F
$+F_y$
At high ε
$A_{xy} = \frac{\partial F_x}{\partial y}$ Positive
$+F_x$

Fig. D.5

force contains a horizontal component (F_x), the static journal locus is rotated through an angle equal to the total force vector angle, to the positions indicated by the dotted crescent curves. The dotted and solid static journal locus on each sub-figure enables the force component changes resulting from positive horizontal displacement perturbations, (a) and (b), and positive vertical displacement perturbations, (c) and (d), to be determined. In a. and b. the horizontal displacements at both high and low ε result in an increase in ε and thus a corresponding increase in F_y. The coefficient Ayx is therefore positive at all ε. Figs. D.5 (c) and (d), however, show that the vertical displacement perturbations correspond to a clockwise shift of the static journal locus at high ε and thus to a positive F_x, but the reverse response is seen to occur at low ε. These figures therefore illustrate the reason why Axy is positive at high ε but negative at low ε.

With regard to Mr. Ng's reference to the kinks seen in Figs. 15 and 21 due to load vector angle (θ_f) variation, this section of the parameter variation study has only been done for $\varepsilon = .7$. Since the kinks result from the interference of an oil groove with the hydrodynamic action, these kinks are expected to occur at other values of ε. The severity of the kinks should diminish with decreasing ε due to the corresponding reduction in the effectiveness of the hydrodynamic action.

Mr. Ng was quite correct in noting that the kinks in Figs. 15 and 21 occurred at $\theta_f = 27°$ and not 37° as stated in the paper.

I must also express my appreciation to Mr. Ng for spotting the errors in the expressions for dimensionless damping coefficient in APPENDIX 1. Originally I used expressions based on journal surface velocity u and later changed to the use of angular velocity ω. The error occurred during this transformation, and the expressions for $\overline{B}$ should be corrected by deleting the D from the denominator. The data given in Figs. 19 and 20 is valid for the correct dimensionless damping coefficients.

Mr. Ng asked for a curve of $\overline{\alpha} = 0$ in Fig. 9. This would be a straight line coincident with the horizontal axis, since zero misalignment results in zero oil film moment!

I have not carried out any work on thrust bearings, and suggest reference (D4) as a possible source of data for dynamic coefficients.

Referring to the Society's acceptance of L/D=2 for water lubricated sterntube bearings designed for hydrodynamic operation, I am not aware of any related service experience. Water lubricated bearings undoubtedly have the advantage of simplicity, but I do not share Mr. Ng's optimism for any future expansion of their utilisation. The fundamental disadvantage is the much lower viscosity of water in relation to oil. This will continue to prevent the attainment of hydrodynamic lubrication in all but lightly loaded and higher shaft speed applications. The elimination of axial grooves from the bottom half of the bearing would help to promote hydrodynamic lubrication, but the problem of a low viscosity lubricant remains. This situation may be quantified approximately by use of Fig. 27 assuming a viscosity of about 5.10^{-4}Pa.s. for water.

ADDITIONAL REFERENCES

(D1) S. HYAKUTAKE, R. ASAI, M. INOUE, K. FUKAHORI, N. WATANABE and M. NONAKE. "Measurement of Relative Displacement between Sterntube Bearing and Shaft on a 210,000 d.w.t. Tanker". Japan Shipbuilding and Marine Eng. Vol. 7, No. 1, 1973 – pp. 32–40.

(D2) R. W. JAKEMAN. "Performance and Oil Film Dynamic Coefficients of a Misaligned Sterntube Bearing". ASLE Transactions, Vol. 29, No. 4, Oct 1986 – pp. 441–450.

(D3) J. W. LUND and K. K. THOMSEN. "A Calculation Method and Data for the Dynamic Coefficients of Oil Lubricated Journal Bearings". Publication unknown.

(D4) L. VASSILOPOULOS. "Methods for Computing Stiffness and Damping Properties of Main Propulsion Thrust Bearings". International Shipbuilding Progress, Vol. 29, No. 329, Jan 1982 – pp. 13–31.

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Lloyd's Register Technical Association

MARPOL 73/78—ANNEX II CONTROL OF POLLUTION BY NOXIOUS LIQUID SUBSTANCES IN BULK

C. M. Magill

Paper No. 3. Session 1986–87

FOR PRIVATE CIRCULATION AMONGST THE STAFF ONLY

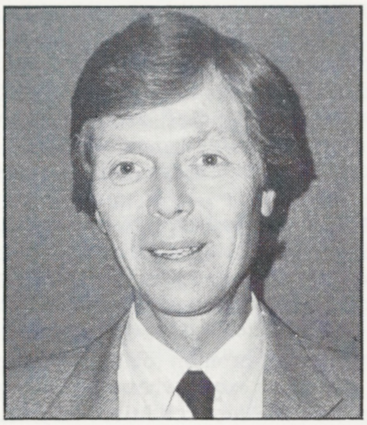
Written contributions to the discussion of this paper are invited from members of the Lloyd's Register Technical Association. To ensure inclusion in the discussion paper, the contributions should be received by the Hon. Secretary in London not later than the 31st March, 1987.

Hon. Sec. C. M. Magill
71 Fenchurch Street, London, EC3M 4BS

MARPOL 73/78—ANNEX II
CONTROL OF POLLUTION BY NOXIOUS LIQUID SUBSTANCES IN BULK

by

C. M. Magill



C. M. Magill began his career as a student apprentice at Harland & Wolff Ltd. and the University of Newcastle upon Tyne, subsequently moving to Stone Manganese Marine Ltd. to work on propeller design. He then spent 5 years with the Royal National Lifeboat Institution where he was involved in the design and development of the Arun and Thames Class fast offshore lifeboats and obtained a Yacht Masters qualification in order to carry out evaluation trials at sea.

In 1975 he took up duties with Lloyd's Register in Glasgow and mainly worked offshore on the Total Frigg and Shell Brent construction projects through to first oil. He was transferred to the International Conventions Department, H.Q., in 1979 and since then has been involved in plan approval for IMO Chemical and Gas Carrier Codes; pollution prevention (oil and chemicals); and fire protection, detection and extinction aspects of ships, fixed and mobile offshore installations, diving systems and land based oil storage terminals.

TABLE OF CONTENTS

1. INTRODUCTION

The International Convention for the Prevention of Pollution from Ships, 1973, as modified by the Protocol of 1978 (together called MARPOL 73/78) has five annexes dealing with various forms of pollution of the sea. These are:

Annex I	Regulations for the Prevention of Pollution by Oil.
Annex II	Regulations for the Control of Pollution by Noxious Liquid Substances in Bulk.
Annex III	Regulations for the Prevention of Pollution by Harmful Substances Carried by Sea in Packaged Forms.
Annex IV	Regulations for the Prevention of Pollution by Sewage from Ships.
Annex V	Regulations for the Prevention of Pollution by Garbage from Ships.

Signatories to the Convention are bound to accept Annexes I and II but not necessarily Annexes III, IV and V which are 'optional'.

Until now, most activity has been related to Annex I, the prevention of pollution by oil, the implementation date of which was separately dealt with by the 1978 Protocol in order to reduce the delay being experienced in achieving entry into force of the complete Convention. One of the significant reasons for the delay was associated with Annex II and the difficulty being experienced in developing procedures whereby ships could achieve the specified maximum concentration of discharges of noxious liquid substances in the wake astern of a ship without having to transfer large quantities of water/residue mixtures to shore reception facilities when in port. These matters have now been resolved and the Society's activity in matters relating to Annex II have been increasing rapidly. It is now important to indicate on all communications which annex is being referred to.

Two guidance documents, one for new ships and one for existing ships, have been prepared by the International Conventions Department to assist shipowners and shipbuilders in achieving compliance with the requirements of Annex II. Copies of these documents have also been circulated to all ports under cover of ICD/ICL 182. This paper is intended only to be a surveyors ready reference supplement to the subject and reference should be made to the Regulations and/or the guidance circulated with the ICD/ICL should more detailed information on any particular aspect be required.

2. REGULATION DEVELOPMENT

It is generally acknowledged that the specialised chemical tanker trade is characterised by high standards of design, construction, maintenance and equipment, coupled with good management by the operators and crews. The excellent safety record of chemical tankers and the absence of major disasters involving large scale pollution from dangerous chemicals must be, in no small measure, due to these factors. The main thrust of the new pollution control measures introduced by MARPOL 73/78—Annex II is therefore on the prevention of operational

pollution resulting from tank cleaning after cargo discharge.

The Code for the Construction and Equipment of Ships Carrying Dangerous Chemicals in Bulk (Bulk Chemical (BCH) Code), adopted by IMO in 1971, was the first international safety standard developed for vessels carrying dangerous chemicals in bulk and, among other things, took into account pollution as a result of damage, but not operational pollution. The Code has been updated and amended in the light of experience over the intervening years, the 10th set of amendments currently being applicable and the 11th set being considered by IMO. Compliance with the Code is not mandatory but has been necessary for ships of some flags as a condition of registration and for ships of all flags in order to trade in certain parts of the world, notably the U.S.A. However, this is about to change since, under MARPOL 73/78—Annex II, compliance with a code is mandatory.

Annex II of MARPOL 73/78 will enter into force on the 6th April 1987 and is applicable to ships carrying any of the noxious liquid substances in bulk that are categorised and listed in Appendix II of the Annex. In addition to the operational pollution prevention aspects, Regulation 13 requires chemical tankers carrying substances of pollution category A, B and C to comply with the provisions of the BCH Code in the case of existing ships (keels laid before 1st July 1986) or the International Code for the Construction and Equipment of Ships carrying Dangerous Chemicals in Bulk (International Bulk Chemical (IBC) Code) in the case of new ships (keels laid on or after 1st July 1986).

This definition date for new and existing ships is related to the entry into force date of another Convention, the 1983 Amendments to the International Convention for the Safety of Life at Sea, 1974 (SOLAS '74). Under the provisions of the 1983 Amendments, it will be mandatory for new chemical tankers (keels laid on or after 1st July 1986) to comply with the IBC Code. Thus two intimately related Conventions are now applicable to chemical tankers, MARPOL mainly for operational pollution prevention aspects and SOLAS for safety aspects.

At IMO, the control of regulation development concerning pollution prevention is exercised by the Marine Environment Protection Committee (MEPC) and safety matters by the Maritime Safety Committee (MSC). Recognising the special relationship between the two Conventions, and to enable only one certificate embracing both safety and pollution prevention aspects to be issued, MEPC and MSC have agreed to harmonise their respective requirements.

For the MARPOL Convention, MEPC has adopted the safety requirements of the BCH and IBC Codes as revised and expanded by the pollution prevention aspects. The consolidated regulations were adopted on the 5th December 1985 and are contained in Resolutions MEPC 20(22) and MEPC 19(22) for the revised BCH and IBC Codes respectively. Unfortunately the amendment procedure required by the SOLAS Convention is such that MSC cannot progress the adoption of the pollution prevention aspects into the IBC code until after the entry into force on 1st July 1986 of the 1983 Amendments to SOLAS '74 and this will take some time. However, since the amended IBC Code as adopted by MEPC contains all the required safety provisions for SOLAS purposes, it has been agreed that only one certificate covering both aspects need be issued even though formal harmonisation has not yet been achieved.

This rather complex situation can be summarised by saying that Certificates of Fitness issued under the provisions of either the BCH or IBC Codes on or after 6th April 1987, will include the operational pollution prevention aspects of MARPOL 73/78—Annex II. Ships carrying noxious liquid substances in bulk which have been assigned a pollution category, but are not subject to the special safety provisions of either the BCH or IBC Codes, will be required to comply only with the operational pollution prevention aspects of MARPOL 73/78—Annex II and be issued with an International Pollution Prevention Certificate for the Carriage of Noxious Liquid Substance in Bulk (NLS Certificate).

Where a flag administration is not a party to both SOLAS '74 and MARPOL 73/78, BCH and IBC Code Certificates of Fitness incorporating MARPOL aspects cannot be issued. In such cases the MEPC has indicated that the port state authorities should give appropriate recognition to documents such as Certificates of Compliance. Similar provisions will apply to NLS Certificates of Compliance when issued to ships flying the flags of States which are not parties to MARPOL 73/78.

The principle that oil and water do not mix and are therefore easily separated is used by Annex I for the load-on-top procedures with the controlled discharge of dirty water ballast being required to be above the waterline for the purpose of observation. However, because most chemical substances will mix with water and are not easily separated from it, the main principle of Annex II is to dilute cargo residues in sea water to prescribed limits depending on their pollution hazard and facilitate the distribution of discharges by utilising the wake of the ship. Thus Annex II discharges are required to be made below the waterline and in such a way that the water/residue mixture is retained in the ship's boundary layer and carried aft when en route to be distributed by the wake astern.

The Annex II regulations prohibit the discharge into the sea of noxious liquid substances except when the discharge is made under specified conditions. These conditions vary according to the degree of hazard a noxious liquid substance poses to the marine environment. For this purpose noxious liquid substances have been divided into four categories, A, B, C and D, with the following definitions for each category:

(i) Category A—Noxious liquid substances which if discharged into the sea from tank cleaning or deballasting operations would present a major hazard to either marine resources or human health or cause serious harm to amenities or other legitimate uses of the sea and therefore justify the application of stringent anti-pollution measures.

(ii) Category B—Noxious liquid substances which if discharged into the sea from tank cleaning or deballasting operations would present a hazard to either marine resources or human health or cause harm to amenities or other legitimate uses of the sea and therefore justify the application of special anti-pollution measures.

(iii) Category C—Noxious liquid substances which if discharged into the sea from tank cleaning or deballasting operations would present a minor hazard to either marine resources or human health or cause minor harm to amenities or other legitimate uses of the sea and therefore require special operational conditions.

(iv) Category D—Noxious liquid substances which if discharged into the sea from tank cleaning or deballasting operations would present a recognizable hazard to either marine resources or human health or cause minimal harm to amenities or other legitimate uses of the sea and therefore require some attention in operational conditions.

Guidelines for use in assessing and categorising noxious liquid substances have been developed by IMO but are not included in this paper since they are primarily a matter relevant only to the manufacturers of the substances and the flag administrations of the loading/unloading ports and the carriers.

Regulation 5 of Annex II specifies the conditions under which discharge of residues of categories A, B, C and D substances may be made. These conditions, which are detailed later, include such parameters as the maximum quantity which may be discharged into the sea; the speed of the ship; the distance from the nearest land; the depth of water; the maximum concentration of substance in the ship's wake or dilution of

substance prior to discharge. Other factors which influence the conditions of discharge of category B and C substances are whether the substance has a high or low viscosity at the unloading temperature and whether the substance is solidifying or non-solidifying at the unloading temperature. Also, in the case of category B substances carried in some 'existing' ships, whether the substance is miscible or immiscible in water. In certain sea areas, referred to as 'Special Areas', more stringent discharge criteria apply. The designated special areas are the Baltic and the Black Sea. It should be noted that the special areas designated in the regulations of Annex I are not the same as those in Annex II.

The "procedures and arrangements for discharge" referred to in Regulations 5, 5A and 8 of Annex II are the requirements developed by IMO for the uniform guidance of approving authorities and are contained in the Standards for Procedures and Arrangements for the Discharge of Noxious Liquid Substances (Resolution MEPC 18(22) adopted on 5th December 1985) which are published at the back of the IMO book containing the revised regulations of Annex II.

Regulation 5A of Annex II requires that the efficiency of the cargo pumping system of a tank intended for the carriage of category B or C substances be tested in accordance with 'standards' developed by IMO. Those 'standards' are contained in Resolution MEPC 18(22). The stripping efficiency determined by test is assumed to be the stripping efficiency that can be achieved when unloading the tank in accordance with the same procedures as used for the test.

A simplified synopsis of the basic regulation requirements for both new and existing ships is given in Table 1.

3. CATEGORISATION OF SUBSTANCES

The substances addressed by MARPOL 73/78—Annex II are those listed in the BCH Codes which, together with about 450 other chemical substances known to be shipped in bulk, have been assessed and categorised A, B, C or D in descending order of pollution hazard, or as non-pollutants. Lists of all these substances, sorted into pollution categories, are given in Appendix A of this paper. The lists contain all the substances known to have been categorised at the time of writing but are likely to be increased over the years as carriage of new substances in bulk are proposed.

When it is proposed to carry a substance which has not been categorised then, in the first instance, the Administration of the shipping or manufacturing country should contact IMO to find out if a provisional assessment has already been made for the substance. If no assessment has been made then the Administration of the shipping or manufacturing country should make a

Table 1 MARPOL 73/78—ANNEX II
Basic Requirements

	NEW SHIPS					EXISTING SHIPS				
Pollution Category	A	B	C	D	D(I)	A	B	C	D	D(I)
P & A Manual	×	×	×	×	×	×	×	×	×	×
Cargo Record Book	×	×	×	×	×	×	×	×	×	×
N.L.S. Certificate	—	—	—	×	—	—	—	—	×	—
Code Certificate of Fitness	×	×	×	—	×	×	×	×	—	×
Underwater Discharge	×	×	×	—	—	×	×	×	—	—
Floatability and Damage Stability as required by the codes	×	×	×	—	×	×	×	×	—	×
Stripping residue quantity	—	0.1 m^3	0.3 m^3	—	—	—	0.3 m^3	0.9 m^3	—	—
Alternative residue quantity until 2-10-94 with	—	—	—	—	—	—	1.0 m^3	3.0 m^3	—	—
Controlled discharge rate and	—	—	—	—	—	—	×	—	—	—
Recording device	—	—	—	—	—	—	×	—	—	—

Category D—pollution hazard only
Category D(I)—safety and pollution hazard (cargo in Chemical Codes)

provisional assessment of the substance using the guidance outlined in MARPOL 73/78, Annex II, Appendix I and submit the assessment to the relevant administrations of the loading and unloading ports for their agreement. The assessment should also be given to the ship's flag administration who, subject to their agreement with the assessment, will notify IMO.

Many existing oil tankers carry occasional cargoes of liquid chemicals at present not regulated by the BCH Code. Owners of such ships may well find that these cargoes are now within the scope of MARPOL 73/78—Annex II as chemical pollutants. After this Annex enters into force, some Category D cargoes can continue to be carried without modification to the ship. However, for the carriage of Category A, B, C or D cargoes where compliance with the chemical code is also required then some ship modifications will almost certainly be necessary to achieve full compliance with the provisions of the BCH Code. The capital cost of such modification is unlikely to be commercially viable for the occasional carriage of these cargoes.

For all ships, the conditions of carriage, and discharge of residues of the various substances, are influenced by the pollution category, whether solidifying or not, whether high or low viscosity and, for some existing ships, whether the substance is miscible or immiscible in water. An additional factor is whether the ship is inside or outside a designated special area. The sections following give details of the individual requirements for each category and its governing factors, commencing with the least onerous.

3.1 Assessed Non-Pollutants

These substances are given in Lists E and E1 in Appendix A of this paper.

List E substances are those given in MARPOL 73/78—Annex II, Appendix III and in BCH Code, Chapter VII, or IBC Code, Chapter 18 (i.e. substances to which the chemical codes do not apply). There are no special requirements in respect of MARPOL 73/78—Annex II and, since the chemical codes are not applicable, neither a Certificate of Fitness nor a Certificate for the Carriage of Noxious Liquid Substances in Bulk is necessary.

List E1 substances have no special requirements in respect of MARPOL 73/78—Annex II but are included in the BCH Code, Chapter VI, or IBC Code, Chapter 17, thus compliance with the chemical codes only is necessary.

3.2 Category D Substances

These substances are given in Lists D and D1 in Appendix A of this paper.

List D substances are those assessed as pollution Category D and are also given in BCH Code, Chapter VII, or IBC Code, Chapter 18 (i.e. substances to which the chemical codes do not apply). An NLS Certificate should be issued for the carriage of these substances.

List D1 substances are those assessed as pollution Category D and which also require compliance with the safety provisions of the BCH or IBC Codes. A Certificate of Fitness (incorporating pollution aspects) should be issued for the carriage of these substances.

Residues of Category D substances may be discharged into the sea, above the waterline, provided:

(i) they have been diluted with water to a concentration of one part of cargo to at least ten parts of water.
(ii) the ship is at least 12 miles from the nearest land
(iii) the ship is proceeding en route at not less than 7 knots.

Thus there are no additional hardware requirements for the ship but an approved Procedures and Arrangements (P & A) Manual, discussed later in Section 4, and a Cargo Record Book Book are to be provided.

3.3 Category B and C Substances

These substances are given in Lists B, B1 and C in Appendix A of this paper.

The requirements for the carriage and discharge of residues of Cateogry B and C substances vary depending on a vessels age (new or existing ships), cargo stripping system capabilities and whether in a special area or not.

3.3.1 Stripping System Capabilities and Testing

To carry these categories of cargo, each ship must be equipped with a stripping system capable of achieving the minimum residue quantities in cargo tanks and associated pumping and piping systems after discharge of cargo as specified in Table 1. Verification that a ship can achieve these stripping quantities forms part of the certification procedure and entails water testing of the stripping system for each tank. Guidance for the assessment of residue quantities in cargo tanks, pumps and piping is given in ICD/ICL 166 to which reference should be made before any testing is carried out. It should be noted that the regulations presume that the results of water stripping tests can be repeated in practice for all the cargoes a ships is certified to carry. Thus the stripping procedures used during the water test must be exactly the same procedure given in the ships P & A Manual. The stripping residue quantity measured by test for one tank may be accepted for a similar tank, provided that the structural arrangements of the tanks are similar and the pumping and piping systems are alike in respect of pump arrangements, pipeline lengths, valves, etc. and operating properly.

For example, the minimum number of tanks to be tested for a tank and cargo piping arrangement as shown in Fig. 1 are those indicated by shading. Number 1 Wing Tank is chosen because this has the longest run of the common suction piping, number 8 Wing Tank because it is the largest tank of this type, a slop tank because of the additional piping involved in the cross-over arrangements and number 5 Centre Tank (deepwell pumps in all centre tanks) because this has the longest discharge pipeline to be blown-through "uphill" during the stripping procedures. It should be noted that wing tanks on opposite sides of the ship have been chosen for test in order to prove that both port and starboard pumps are operating satisfactorily.

3.3.2 Underwater Discharge Outlet

Overboard discharge of residue is required to be below the waterline and the discharge outlet designed to be of a diameter large enough to ensure that a jet of the residue/water mixture expelled from it will not pass through the ship's boundary layer when proceeding at not less than 7 knots. The minimum diameter of the underwater discharge outlet is given by:

$$D = \frac{Q_D}{5L}$$

where D = minimum diameter of the discharge outlet (m)

L = distance from the forward perpendicular to the discharge outlet (m)

Q_D = maximum rate selected at which the ship may discharge a residue/water mixture through the outlet (m^3/h)

A smaller diameter than that derived from the above calculation may be acceptable if the discharge outlet is arranged into a sea chest such that the expelled jet of liquid will not be able to penetrate the ship's boundary layer.

For maximum flexibility of use it is recommended that the minimum diameter of outlet be calculated using the maximum nominal discharge rate of the cargo pumps. This will be of great advantage if contaminated ballast water or large amounts of wash water are to be discharged.

The location of the discharge outlet, or outlets, should be arranged to avoid reingestion of cargo residue by the ship's sea

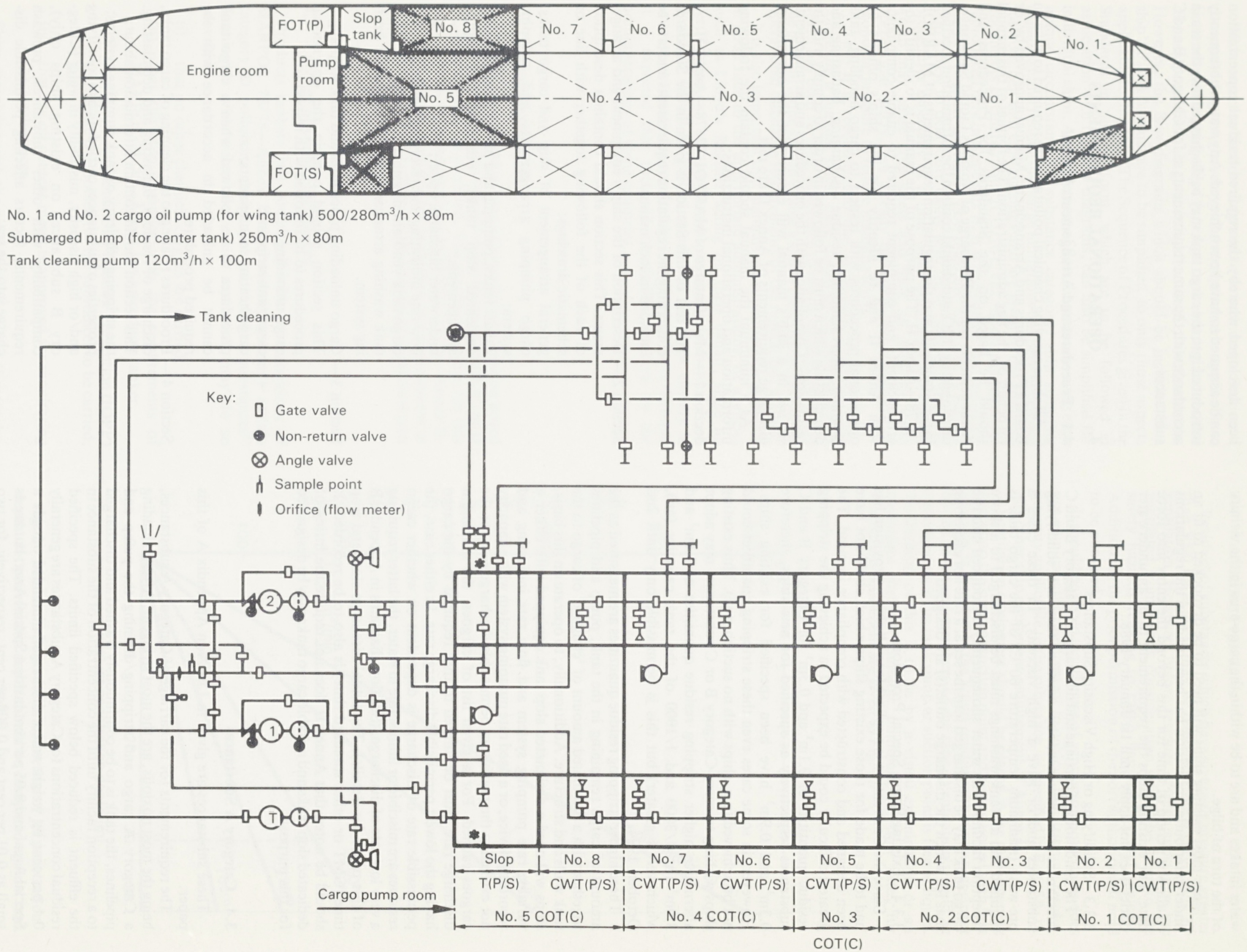


Fig. 1 Arrangement of Tanks and Suction Wells

water intakes and are to be within the cargo area in the vicinity of the turn of bilge.

Since most existing ships will need to be drydocked to fit an underwater outlet, provision has been made in the regulations to allow a period of time for this beyond the entry into force date. For existing ships the requirement to fit an underwater outlet does not apply until 1st January 1988.

3.3.3 Solidifying or High Viscosity Substances

The conditions of carriage and discharge of Category B and C substances are influenced by whether they are solidifying substances and/or have a high viscosity. In these cases a pre-wash (minimum temperature 60°C) of the cargo tank is required and all tank washings must be discharged to a shore reception facility. Any water subsequently introduced into the cargo tank may be discharged into the sea at a rate not exceeding that for which the discharge outlet(s) are designed.

3.3.4 Existing Ships, Special Provisions

Since the Convention is applicable to all ships, provision has had to be made for those existing ships which could not have been designed and constructed with compliance in mind. The greatest difficulty could be expected in achieving the stripping residue quantities of $0.1m^3$ and $0.3m^3$ for Category B and C substances respectively as specified for new ships, therefore $0.3m^3$ and $0.9m^3$ have been specified for existing ships. However, in some cases even these stripping quantities could not be achieved (e.g. ships with no suction well). Thus existing ships wishing to carry Category B or C substances may adopt alternative higher stripping residue quantities of $1.0m^3$ and $3.0m^3$ (or 1/3000 and 1/1000 of the tank capacity (m^3), whichever is greater) but this is permissible only until 2nd October, 1994.

These higher stripping residue quantities are measured as the amount of cargo remaining in the tank, pump and pipelines together with a calculated amount of 'clingage' of cargo to the tank walls and structure. Additionally, to operate on this basis, ships will need a designated slop tank equipped with either a variable rate pumping system and flow rate indicating and recording device, or a fixed rate pumping system with a capacity not exceeding the maximum permissible discharge rate for the cargoes carried. For a given list of cargoes, the maximum discharge rate must be limited to that required by the cargo having the lowest permissible discharge rate. In either case, the permissible rate of discharge is dependent on whether cargo residues are miscible or immiscible in water, the latter requiring a very low rate of discharge. (See flow diagrams in Appendix B of this paper for details). The actual discharge start and stop time (GMT or other standard time) is also to be recorded. It should be noted that Annex II does not require the fitting of dedicated slop tanks and suitable cargo tanks may be designated for this purpose.

3.4 Category A Substances

These substances are given in List A in Appendix A of this paper.

The requirements for the carriage of Category A substances, being the most noxious, are the most stringent. After unloading a Category A cargo and stripping/draining the pump and pipelines, the tank is to be continuously washed and discharged to a reception facility until the concentration of the substance in the effluent is reduced below specified limits. The specified residual concentrations for Category A substances are generally 0.1 per cent by weight where discharge is intended outside a Special Area, and 0.05 per cent inside a Special Area, but are as small as 0.01 per cent and 0.005 per cent, respectively, for two Category A cargoes, carbon disulphide and phosphorus. It is recognised that determination of the residual concentration may be difficult, therefore an alternative prewash procedure has been developed whereby the required residual concentration can be assumed to have been achieved. Any water subsequently introduced into the cargo tank may be discharged into the sea in accordance with the same requirements as for Category B and C substances.

4. OPERATIONAL REQUIREMENTS

4.1 Procedures and Arrangements Manual

All ships carrying chemical pollutants are required to have on board a Procedures and Arrangements Manual approved by, or on behalf of, the flag administration. The format of this manual should be based on the Standards for Procedures and Arrangements for the Discharge of Noxious Liquid Substances developed by IMO and should contain the information specified in the Standards and the regulation requirements of MARPOL 73/78—Annex II. The purpose of the manual is to identify the arrangements and equipment required to enable compliance with Annex II and to identify for the ship's officers all operational procedures with respect to cargo handling, tank cleaning, slops handling, residue discharging, ballasting and deballasting which must be followed in order to comply with the regulations. Compliance with the procedures and arrangements set out in a ship's manual will ensure that the operational discharge requirements of Annex II are met.

As a minimum the manual should contain the following information and operational instructions:

Section 1 — Main features of MARPOL 73/78—Annex II. The text of this Section is given in the Standards concerning the regulation requirements and is to be included in each manual.

Section 2 — Description of the ship's equipment and arrangements. This section should contain a description of each of the following together with line or schematic drawings:
general arrangement of ship and cargo tanks;
cargo pumping arrangements and stripping system;
ballast tanks, pumping and piping arrangements;
dedicated slop tanks, pumping and piping arrangements;
underwater discharge outlet;
flow rate indicating and recording devices;
cargo tank ventilation system;
tank washing arrangements and washwater heating system.

Section 3 — Cargo unloading procedures and tank stripping. This section should contain the operational procedures to be followed with respect to:
cargo unloading;
cargo tank stripping;
cargo heating requirements;
procedures to be followed when a cargo tank cannot be unloaded in accordance with the required procedures.

Section 4 — Procedures relating to the cleaning of cargo tanks, discharge of residues, ballasting and deballasting. This section should contain information about:
the stripping efficiency of tank pumping systems;
properties of substances carried such as solidifying and/or high viscosity, miscibility in water (Category B substances on existing ships only), compatibility with other substances, prewash requirements, factors affecting permissible discharge into the sea;
use of cleaning agents or additives;
use of ventilation procedures for tank cleaning;
flow diagrams of residue disposal procedures.

With respect to the flow diagrams to be included in Section 4, composite diagrams covering all the requirements applicable to both new and existing ships for cargo tank cleaning and residue disposal procedures are given in Addendum A of the Standards. However, the flow diagrams which should be included in a ship's P & A Manual are to be simplified to only those aspects applicable to that ship. The Sequence of Procedures given in the Standards in tabular form, which are to be followed in association with the flow diagrams, can also be simplified for individual ships. A complete set of flow diagrams covering each unique circumstance of substance category, ship stripping capability and residue discharge limitations has been developed for both new and existing ships and are given in the guidance documents circulated with ICD/ICL 182. Some examples are reproduced here in Appendix B. It is considered that use of the Lloyd's Register developed flow diagrams may greatly assist a ship's crew to deal more easily with these complex operating procedures especially at a time when additional workloads, which may cause crew-stress and have an influence upon operational safety, are already on the increase because of economic pressure to reduce manning levels.

Thus it is recommended that great care be taken when compiling a manual since, once the stripping efficiency of a vessel has been verified, adherence to the procedures set out in the manual alone should enable a ship's crew to ensure that the operational discharge requirements of the regulations are met.

It is also important that manuals are submitted for approval in good time to the appropriate administrations, or agencies such as Lloyd's Register acting on their behalf, since no certification can be issued unless an approved manual is on board. It should be kept in mind that, for ships already in service, certification is required by the entry into force date and it is expected that this will be rigidly enforced by the administrations of some countries.

4.2 Cargo Record Book

All ships are required to have a Cargo Record Book on board in which all cargo operations involving Category A, B, C, or D substances are to be recorded. The standard format for this book is given in Appendix IV of MARPOL 73/78—Annex II. The book is to be available for inspection and endorsement at any time by authorised officers of the flag administration or port authority.

4.3 Carriage of Oil-like Substances

Products tankers not complying with the provisions of MARPOL 73/78—Annex II and wishing to carry certain category D pollutants designated as "oil-like substances" (immiscible with water and easily separated) may do so provided:

(i) that no other chemical cargoes are carried,

(ii) that the ship complies with the requirements of MARPOL 73/78—Annex I as a products tanker,

(iii) that the International Oil Pollution Prevention (IOPP) Certificate is endorsed for the substances to be carried, and

(iv) that the oil content meter is approved for monitoring these substances.

Ships complying with the provisions of Annex II may also handle oil-like substances in accordance with Annex I procedures provided these same requirements are complied with.

4.4 Ventilation Procedures

Residues of substances which have a vapour pressure greater than 5000 Pa at 20°C may be removed from a cargo tank by ventilation. However, before residues of such substances are ventilated from a tank consideration should be given to the safety hazards relating to cargo flammability and toxicity. Additionally the prior permission of the port authority will need to be obtained.

It should be noted that the use of ventilation procedures does not eliminate the requirement to achieve the stripping residue quantity applicable to the substance, as specified in Reg. 5A, or any other ship equipment requirement of Annex II.

Ventilation equipment should be placed in the tank opening nearest to the sump or suction point and be positioned such that an airjet is directed at the liquid residue without impingement on the structural members of the tank. Ventilation should be continued until the tank sump or suction point is free of liquid. Verification of this is to be by a visual examination or an equivalent method. Suitable minimum air flow rates for ventilation equipment are given in Fig. 2.

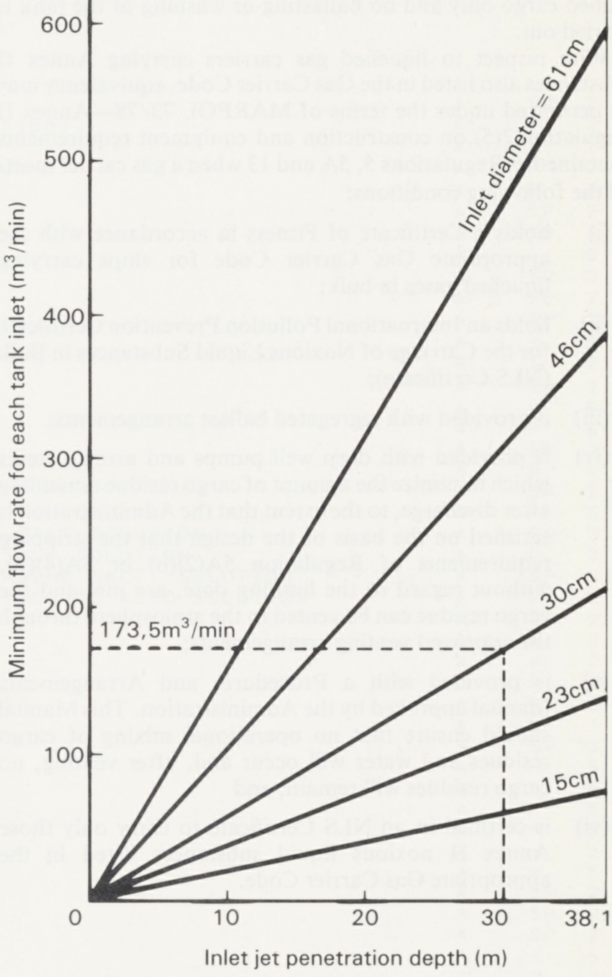


Fig. 2 Minimum flow rate as a function of jet penetration depth. Jet penetration depth should be compared against tank height.

4.5 Prewash Procedures

When a mandatory prewash is specified by the Standards this is to be carried out by means of a rotary water jet. In the case of Category A substances washing machines should be placed in such locations to ensure that all tank surfaces are washed. For Category B and C substances only one location need be used. During washing the amount of water in the tank should be minimised by continuously pumping out slops and promoting flow to the suction point. The number of cycles required and temperature of wash water are specified in Appendix B of the Standards.

5. EXEMPTIONS AND INTERPRETATIONS

Exemption from compliance with the stripping requirements of MARPOL 73/78—Annex II Reg. 5A may be given by the relevant Administrations to dedicated trade ships making restricted voyages between designated ports with adequate reception facilities to receive Category B or C prewash or residue/water mixtures. In such cases no residue is disposed of at sea (see Reg. 5A (6)).

Exemption from compliance with the stripping requirements of Reg. 5A may also be given by the flag Administration for ships where each cargo tank is certified for the carriage of one named cargo only and no ballasting or washing of the tank is carried out.

With respect to liquefied gas carriers carrying Annex II substances also listed in the Gas Carrier Code, equivalency may be permitted under the terms of MARPOL 73/78—Annex II Regulation 2(5) on construction and equipment requirements contained in Regulations 5, 5A and 13 when a gas carrier meets all the following conditions:

(i) holds a Certificate of Fitness in accordance with the appropriate Gas Carrier Code for ships carrying liquefied gases in bulk;

(ii) holds an International Pollution Prevention Certificate for the Carriage of Noxious Liquid Substances in Bulk (NLS Certificate);

(iii) is provided with segregated ballast arrangements;

(iv) is provided with deep well pumps and arrangements which minimize the amount of cargo residue remaining after discharge, to the extent that the Administration is satisfied on the basis of the design that the stripping requirements of Regulation 5A(2)(b) or 5A(4)(b), without regard to the limiting date, are met and the cargo residue can be vented to the atmosphere through the approved venting arrangements;

(v) is provided with a Procedures and Arrangements Manual approved by the Administration. This Manual should ensure that no operational mixing of cargo residues and water will occur and, after venting, no cargo residues will remain; and

(vi) is certified in an NLS Certificate to carry only those Annex II noxious liquid substances listed in the appropriate Gas Carrier Code.

6. CONCLUSION

At first sight the requirements of Annex II may seem to be very onerous, particularly for existing ships not designed or built with compliance in mind. However, there are in fact a number of operational options open to the owner of an existing ship which could mitigate the financial cost involved in ship modification in order to achieve compliance.

Generally the basic requirements for the discharge of residue water mixtures become more onerous as the pollution category increases and this is reflected in the cost. Thus it may well be advantageous to restrict a ship's approved cargo list to only those categories of cargoes carried regularly and to omit from the list all higher category cargoes which have previously been carried only infrequently. Exceptions to this approach would be, for example, some Category A cargoes which do not have hardware requirements in excess of those required for Category B or C substances, although they have much more onerous operational restrictions.

Consideration could also be given to the designation and equipping of specific tanks for the carriage of certain pollution category cargoes where compliance is more easily achieved. For example, centre tanks for Category B substances and below, the majority of wing tanks for Category C and below and perhaps the end wing tanks, or those with similarly complex structural configurations which would cause high residue quantities, for Category D substances or non-pollutants. No doubt other alternatives will be found by the judicious owner to suit individual ships and their trading patterns.

To assist the Owners or Builders of classed vessels the Society has developed a computer program to examine submitted chemical cargo lists, sort the cargoes into pollution categories and define the minimum requirements for their carriage both from safety aspects and pollution prevention aspects. From the output obtained it will be possible to assess the amount of work required to be carried out on any individual ship in order for it to achieve compliance with the requirements of the appropriate chemical Code and MARPOL 73/78—Annex II.

As stated in the introduction, this paper is only intended to be a ready reference supplement concerning Annex II and reference should be made to the Regulations and the guidance documents associated with ICD/ICL's 166 and 182 in all cases where more detailed information is required.

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APPENDIX A

CARGO LISTS

The categorisation of cargoes is under constant review at IMO and the following lists are based on the most up to date information available at the time of publication.

LEGEND

The symbols shown in the three columns at the right hand side of the cargo lists are as follows:

1st Column		**POLLUTION CATEGORY**
III	—	Cargoes assessed as non-pollutants and listed in Annex II Appendix III or BCH/IBC Code Ch.VI/17 Col. c.
A, B, C or D	—	Cargoes listed in Annex II Appendix II or BCH/IBC Code Ch.VI/17 Col. c.
DO or CO	—	Cat D & C cargoes indentified by IMO as being oil like substances according to the Unified Interpretations.
BM	—	Cat B cargoes indentified by IMO as being miscible in water.
B?	—	Cat B cargoes not yet identified by IMO as being miscible/immiscible.

(Note that the M and the ? have no significance for NEW SHIPS)

2nd Column		**HAZARD DESIGNATION**
S	—	Cargo has a safety hazard.
P	—	Cargo has a pollution hazard.
S/P	—	Cargo has a safety and pollution hazard.

3rd Column		**SHIP TYPE**
1, 2 or 3	—	Ship type according to BCH/IBC Code Chapter VI/17 Col c.
4	—	Cargo is listed in BCH/IBC Code Chapter VII/18.
5	—	Cargo not listed in BCH/IBC Code.

List A

Pollution Category A in CH. VI/17 of BCH/IBC Codes

ACETONE CYANOHYDRIN	A	SP	2
ANTHRACENE OIL (coal tar fraction)	A	P	2
BUTYL BENZYL PHTHALATE	A	P	2
CALCIUM BROMIDE / ZINC BROMIDE MIXTURES SOLUTION	A	P	2
CALCIUM NAPHTHENATE IN MINERAL OIL	A	P	3
CARBOLIC OIL	A	SP	2
CARBON DISULPHIDE	A	SP	2
1-CHLOROHEPTANE	A	-	5
o-CHLOROTOLUENE	A	SP	3
CHLOROTOLUENE (mixed isomers)	A	SP	2
COAL TAR	A	SP	2
COBALT NAPHTHENATE IN SOLVENT NAPHTHA	A	-	5
CREOSOTE (wood)	A	SP	2
CRESOLS, (mixed isomers)	A	SP	2
CRESYL DIPHENYL PHOSPHATE	A	-	5
CRESYLIC ACID	A	-	5
DECYL ACRYLATE	A	SP	2
DIBUTYL PHTHALATE	A	P	2
2,4-DICHLOROPHENOL	A	SP	2
2,4-DICHLOROPHENOXY-ACETIC ACID	A	-	5
2,4-DICHLOROPHENOXYACETIC ACID, DIETHANOLAMINE SALT SOLUTION	A	SP	3
2,4-DICHLOROPHENOXYACETIC ACID, DIMETHYLAMINE SALT SOLUTION	A	SP	3
2,4-DICHLOROPHENOXYACETIC ACID, TRIISOPROPANOLAMINE SALT, SOLUTION	A	SP	3
DIETHYLAMINE SALT OF 4-CHLORO-2-METHYLPHER NOXYACETIC ACID SOLUTION	A	P	3
DIISOPROPYLBENZENE (all isomers)	A	P	2
DIPHENYL	A	P	2
DIPHENYL/DIPHENYL OXIDE MIXTURES	A	P	2
DIPHENYL ETHER	A	P	3
DIPHENYL OXIDE / DIPHENYL PHENYL ETHER MIXTURES	A	P	3
DODECYLPHENOL	A	P	1
o-ETHYLPHENOL	A	SP	3
IRON CHLORIDE, COPPER CHLORIDE SOLUTION	A	-	5
ISODECYL ACRYLATE	A	-	5
METHANETHIOL	A	-	5
METHYL NAPHTHALENE	A	-	5
alpha-METHYLNAPHTHALENE	A	-	5
beta-METHYLNAPHTHALENE	A	-	5
alpha-METHYLSTYRENE	A	SP	2
MOTOR FUEL ANTI-KNOCK COMPOUNDS	A	SP	2
NAPHTHALENE, (molten)	A	SP	2
NAPHTHENIC ACIDS	A	-	5
NONYLPHENOL	A	P	2
PHOSPHORUS, YELLOW OR WHITE	A	SP	1
PINENE	A	P	3
PINE OIL	A	P	3
ROSIN	A	P	3
TALL OIL, CRUDE AND DISTILLED	A	P	3
TRICRESYL PHOSPHATE CONTAINING 1% OR MORE ORTHO ISOMER	A	SP	1
TRICRESYL PHOSPHATE CONTAINING LESS THAN 1% ORTHO ISOMER	A	P	2
TRIETHYLBENZENE	A	P	2
TRIXYLYL PHOSPHATE	A	P	1
VINYL TOLUENE	A	SP	3

List B

Pollution Category B in CH.VI/17 of BCH/IBC Codes

ACRYLONITRILE	B	SP	2
ALCOHOLS, C10,C11,C12 as individuals & mixtures	B?	–	5
ALCOHOLS, (C13,C15) POLY (3-11) ETHOXYLATES	B?	–	5
ALKYL BENZENE SULPHONATE (branched chain)	B?	–	5
ALLYL ALCOHOL	BM	SP	2
ALLYL CHLORIDE	B	SP	2
AMMONIUM SULPHIDE SOLUTION (45% or less)	BM	SP	2
BENZYL CHLORIDE	B	SP	2
n-BUTYL BUTYRATE	B?	–	5
n-BUTYRALDEHYDE	B	SP	3
BUTYRIC ACID	BM	SP	3
CALCIUM HYPOCHLORITE SOLUTION (more than 13%)	B?	SP	3
CAMPHOR OIL	B	SP	2
CARBON TETRACHLORIDE	B	SP	3
CHLOROBENZENE	B	SP	2
CHLOROFORM	B	SP	3
m-CHLOROTOLUENE	B	SP	3
p-CHLOROTOLUENE	B	SP	2
COAL TAR NAPHTHA SOLVENT	B	SP	3
CROTONALDEHYDE	B	SP	2
n-DECALDEHYDE	B?	–	5
DECENE	B	P	3
DECYL ALCOHOL (all isomers)	B	P	3
m-DICHLOROBENZENE	B?	–	5
o-DICHLOROBENZENE	B	SP	2
p-DICHLOROBENZENE (molten)	B?	SP	2
1,1-DICHLOROETHANE	B	SP	3
DICHLOROETHYL ETHER	B	SP	2
1,6-DICHLOROHEXANE	B?	–	5
1,1-DICHLOROPROPANE	B?	–	5
1,2-DICHLOROPROPANE	B	SP	2
1,3-DICHLOROPROPANE	B	SP	2
1,3-DICHLOROPROPENE	B	SP	2
DICHLOROPROPENE/DICHLOROPROPANE MIXTURES	B	SP	2
DICHLOROPROPYL ETHER	B?	–	5
DIETHYL SULPHATE	B	SP	2
DIGLYCIDYL ETHER OF BISPHENOL-A	B?	P	3
DIISOBUTYLENE	B	P	3
DIISOBUTYL PHTHALATE	B	P	3
N,N-DIMETHYLACETAMIDE	B?	–	3
DIPHENYLMETHANE DIISOCYANATE	B?	SP	2
DIPHENYLOL PROPANE-EPICHLOROHYDRIN RESINS	B?	P	3
DODECENE (all isomers)	B	P	3
ETHYL ACRYLATE	B	SP	2
ETHYLENE DIBROMIDE	B	SP	2
ETHYLENE DICHLORIDE	B?	SP	2
ETHYLENEIMINE	B	–	5
2-ETHYLHEXYLAMINE	B	SP	2
ETHYLIDENE NORBORNENE	B	SP	3
2-ETHYL-3-PROPYLACROLEIN	B	SP	3
ETHYLTOLUENE	B?	P	3
FATTY ALCOHOLS (C12-C20)	B	P	3
FUMARIC ADDUCT OF ROSIN WATER DISPERSION	BM	P	3
GLYCIDYL ESTER OF C10 TRIALKYL ACETIC ACID	B?	P	3
HEPTYL ACETATE	B	P	3
HEXYL ACETATE	B?	P	3
2-HYDROXYETHYL ACRYLATE	BM	SP	2
ISOBUTYL ISOBUTYRATE	B?	P	3
ISOPHORONE DIISOCYANATE	B?	SP	3
ISOPROPYLBENZENE	B	P	3
LACTONITRILE SOLUTION (80% or less)	B?	SP	2
MERCAPTOBENZOTHIAZOL SODIUM SALT SOLUTION	B?	SP	3
METHACRYLONITRILE	B	SP	2
METHYL BENZONATE	B?	–	5
METHYLENE DIANILINE & HIGHER MOLE. WEIGHT POLYMERS/ o-DICHLOROBENZENE MIX	B?	–	5
2-METHYL-5-ETHYLPYRIDINE	B	SP	3
METHYL HEPTYL KETONE	B	P	3
2-METHYLPYRIDINE	BM	SP	2
3-METHYLPYRIDINE	B?	SP	2
4-METHYLPYRIDINE	BM	SP	2
N-METHYL-2-PYRROLIDONE	BM	P	3
METHYL SALICYLATE	B	P	3
NEODECANOIC ACID	B	P	3
NITROBENZENE	B	SP	2
NONENE	B	P	3
NONYLPHENOL POLY (4-12) ETHOXYLATES	B?	–	5
OCTENE, (all isomers)	B	P	3
OCTYL ALDEHYDES	B	P	3
OCTYL NITRATE, (all isomers)	B	P	3
OLEFINS, straight chain mixtures	B	P	3

(continued)

OLEFINS (C6-C8 mixtures)	B?	-	5
alpha-OLEFINS (C6-C18 mixtures)	B	P	3
PENTACHLOROETHANE	B	SP	2
PERCHLOROETHYLENE	B	SP	3
n-PROPYL CHLORIDE	B?	-	5
PROPYLENE TRIMER	B	P	3
PYRIDINE	BM	SP	3
ROSIN SOAP (disproportionated) SOLUTION	BM	P	3
SODIUM DICHROMATE SOLUTION (70% or less)	BM	SP	2
SODIUM HYDROSULPHIDE SOLUTION, (45% or less)	BM	SP	3
SODIUM HYDROSULPHIDE / AMMONIUM SULPHIDE SOLUTION MIXTURE	BM	SP	2
SODIUM NITRATE SOLUTION	B?	-	5
SODIUM SULPHIDE SOLUTION	B?	-	5
SODIUM SULPHONATE	B?	-	5
SODIUM THIOCYANATE	B?	-	5
SODIUM THIOCYANATE (56% or less)	B?	P	3
STYRENE MONOMER	B	SP	3
SULPHUR CHLORIDE	B?	-	5
TALL OIL (disproportionated) SOLUTION	BM	P	3
TETRACHLOROETHANE	BM	SP	3
TRIBUTYL PHOSPHATE	B?	P	3
1,1,1-TRICHLOROETHANE	B	P	3
1,1,2-TRICHLOROETHANE	B	SP	3
TRICHLOROETHYLENE	B	SP	3
1,2,3-TRICHLOROPROPANE	B	SP	2
1,2,3-TRIMETHYLBENZENE	B?		5
1,2,4-TRIMETHYLBENZENE	B?	P	3
1,3,5-TRIMETHYLBENZENE	B?		5
TRIMETHYLHEXAMETHYLENE DIISOCYANATE (2,2,4-and-2,4,4-ISOMERS)	BM	SP	2
TURPENTINE	B	P	3
1-UNDECENE	B?	P	3
VINYLIDENE CHLORIDE	B	SP	2
WHITE SPIRIT, low (15-20%) aromatic	B	P	2

List B1

Pollution Category B with high melting point

o-CHLORONITROBENZENE	B	SP	2
DINITROTOLUENE (molten)	B	SP	2
DODECYL ALCOHOL	B	P	3
DODECYL DIPHENYL OXIDE DISULFONATE SOLUTION	BM	SP	3
o-NITROPHENOL (molten)	B	SP	2
PHENOL	B	SP	2
1,2,4-TRICHLOROBENZENE	B	SP	2
UNDECYL ALCOHOL	B	P	3
XYLENOLS	B	SP	3

List C

Pollution Category C in CH. VI/17 of BCH/IBC Codes

ACETALDEHYDE	C	-	5
ACETIC ACID	C	SP	3
ACETIC ANHYDRIDE	C	SP	2
ACETYL CHLORIDE	C	-	5
ALCOHOLS, C7,C8,C9 as individuals & mixtures	C	-	5
ALKYL ACRYLATE VINYL PYRIDINE COPOLYMER IN TOLUENE	C	P	3
ALKYLAMINE MIXTURES	C	-	5
ALKYL BENZENE SULPHONATE (straight chain)	C	-	5
ALKYL BENZENE SULPHONIC ACID	C	SP	3
AMMONIA AQUEOUS, (28% or less)	C	SP	3
AMMONIUM THIOSULPHATE SOLUTION (60% or less)	C	SP	3
n-AMYL ACETATE	C	P	3
sec-AMYL ACETATE	C	P	3
AMYL ACETATE,commercial	C	P	3
ANILINE	C	SP	2
BENZALDEHYDE	C	-	5
BENZENE AND MIXTURES HAVING 10% BENZENE or more	C	SP	3
BENZYL ACETATE	C	P	5
BENZYL ALCOHOL	C	P	3
n-BUTYL ACETATE	C	P	3
BUTYLAMINE (all isomers)	C	SP	2
BUTYL BENZENES	C	P	3
1,2-BUTYLENE OXIDE	C	SP	3
n-BUTYL ETHER	C	SP	3

(continued)

Pollution Category C in CH. VI/17 of BCH/IBC Codes *(continued)*

BUTYL HEPTYL KETONE		C	P	3
CALCIUM HYPOCHLORITE SOLUTION	(13% or less)	C	SP	3
CHLOROACETIC ACID (80% or less)		C	SP	2
CHLOROACETONE		C	-	5
4-CHLORO-2-METHYLPHENOXYACETIC ACID, DIMETHYLAMINE SALT SOLUTION		C	P	3
2- or 3-CHLOROPROPIONIC ACID		C	P	3
CHLOROSULPHONIC ACID		C	SP	1
CREOSOTE (coal tar)		C	SP	3
CYCLOHEXANE		CO	P	3
CYCLOHEXANE / CYCLOHEXANOL MIXTURE		C	-	5
CYCLOHEXANOL		C	P	3
CYCLOHEXYLAMINE		C	SP	3
p-CYMENE		CO	P	3
DIBENZYL ETHER		C	-	5
DIBUTYLAMINE		C	SP	3
2,2-DICHLOROISOPROPYL ETHER		C	SP	2
DIETHYLAMINE		C	SP	3
DIETHYLAMINOETHANOL		C	SP	3
DIETHYLBENZENE		CO	P	3
DIETHYLENE GLYCOL METHYL ETHER		C	P	3
DI-(2-ETHYLHEXYL) PHOSPHORIC ACID		C	SP	3
DIETHYL MALONATE		C	-	5
DIETHYL PHTHALATE		C	P	3
DIISOBUTYLAMINE		C	SP	2
DIISOPROPANOLAMINE		C	SP	3
DIISOPROPYLAMINE		C	SP	2
DIMETHYLAMINE SOLUTION	(45% or less)	C	SP	3
DIMETHYLAMINE AQUEOUS,	(greater than 45% but not greater than 55%)	C	SP	2
DIMETHYLAMINE AQUEOUS,	(greater than 55% but not greater than 65%)	C	SP	2
N,N-DIMETHYLCYCLOHEXYLAMINE		C	SP	2
DIMETHYL PHTHALATE		C	P	3
DIPENTENE		CO	P	3
DI-n-PROPYLAMINE		C	SP	3
DODECYLAMINE, TETRADECYLAMINE MIXTURE		C	SP	3
DODECYL BENZENE		CO	P	3
EPICHLOROHYDRIN		C	SP	2
2-ETHOXYETHYL ACETATE		C	P	3
ETHYLAMINE		C	SP	2
ETHYLAMINE SOLUTIONS,	(72% or less)	C	SP	2
ETHYL AMYL KETONE		C	-	5
ETHYLBENZENE		CO	P	3
N-ETHYLBUTYLAMINE		C	SP	3
ETHYL BUTYLATE		C	P	3
ETHYLENE CHLOROHYDRIN		C	SP	2
ETHYLENEDIAMINE		C	SP	2
ETHYLENE GLYCOL DIACETATE		C	P	3
ETHYLENE GLYCOL ETHYL ETHER ACETATE		C	P	3
ETHYL HEXYL PHTHALATE		C	P	3
FERRIC CHLORIDE SOLUTION		C	-	5
FORMALDEHYDE SOLUTIONS	(45% or less)	C	SP	3
FURFURAL		C	SP	3
FURFURYL ALCOHOL		C	P	3
HEPTANOL (all isomers)		C	P	3
HEPTENE (mixed isomers)		CO	P	3
HEXAHYDROCYMENE		C	-	5
HEXAMETHYLENEDIAMINE SOLUTIONS		C	SP	3
HEXAMETHYLENEIMINE		C	SP	2
1-HEXENE		CO	P	3
2-HEXENE		CO	P	3
HYDROGEN PEROXIDE SOLUTIONS,	(over 8% but not over 60%)	C	SP	3
HYDROGEN PEROXIDE SOLUTIONS,	(over 60% but not over 70%)	C	SP	2
ISOAMYL ACETATE		C	P	3
ISOBUTYL ACETATE		C	P	3
ISOBUTYL FORMATE / ISOBUTANOL MIXTURES		C	-	5
ISOBUTYRALDEHYDE		C	SP	3
ISODECALDEHYDE		C	-	5
ISOPRENE		C	SP	3
ISOPROPANOLAMINE		C	SP	3
ISOPROPYLAMINE		C	SP	2
ISOVALERALDEHYDE		C	SP	3
MAGNESIUM SULPHONATE		C	P	3
METHYL ACRYLATE		C	SP	2
METHYLAMINE SOLUTIONS	(42% or less)	C	SP	2
METHYLAMYL ACETATE		C	P	3
METHYLAMYL ALCOHOL		C	P	3
METHYL AMYL KETONE		C	P	3
2-METHYL BUTYRALDEHYDE		C	-	5
METHYL BUTYLATE		C	P	3
METHYLETHANOLAMINE		C	-	5
2-METHYL-6-ETHYL ANILINE		C	SP	3

(continued)

Pollution Category C in CH. VI/17 of BCH/IBC Codes *(continued)*

2-METHYL-1-PENTENE		CO	P	3
NITRATING ACID (mixture of sulphuric and nitric acids)		C	P	2
NITRIC ACID (less than 70%)		C	-	2
NITRIC ACID (70% and over)		C	S	2
o- and p-NITROTOLUENE		C	SP	2
NONYL ALCOHOL		C	P	3
OCTANOL, (all isomers)		C	P	3
OLEUM		C	SP	2
PARALDEHYDE		C	SP	3
1,3-PENTADIENE		C	SP	3
n-PENTANE		CO	P	3
PENTENE (all isomers)		CO	P	3
1-PHENYL-1-XYLYL ETHANE		CO	P	3
PHTHALIC ANHYDRIDE		C	SP	3
POLYETHYLENE POLYAMINES		C	SP	3
POTASSIUM HYDROXIDE SOLUTION		C	SP	3
n-PROPANOLAMINE		C	SP	3
PROPIONIC ANHYDRIDE		C	SP	3
PROPIONITRILE		C	SP	2
n-PROPYLAMINE		C	SP	2
n-PROPYL BENZENE		C	-	5
PROPYLENE DIMER		CO	P	3
SODIUM ALUMINATE SOLUTION		C	-	5
SODIUM BOROHYDRIDE (15% OR LESS) / SODIUM HYDROXIDE SOLUTION		C	SP	3
SODIUM HYPOCHLORITE SOLUTION,	(15% or less)	C	SP	3
SODIUM SULPHITE SOLUTION		C	-	5
SULPHURIC ACID		C	SP	3
SULPHURIC ACID, SPENT		C	SP	3
SULPHUROUS ACID		C	-	5
TALL OIL FATTY ACID	(resin acids less than 20%)	C	P	3
TANNIC ACID		C	-	5
TETRADECYLBENZENE		C	P	3
TETRAHYDRONAPHTHALENE		CO	P	3
1,2,3,5-TETRAMETHYLBENZENE		C	-	5
TOLUENE		CO	P	3
TOLUENEDIAMINE		C	SP	2
TOLUENE DIISOCYANATE		C	SP	2
o-TOLUIDINE		C	SP	2
1,1,2-TRICHLORO -1,2,2- TRIFLURORETHANE		C	P	3
TRIDECYLBENZENE		C	P	3
TRIETHYLAMINE		C	SP	2
TRIMETHYLAMINE		C	-	5
2,2,4-TRIMETHYL-1,3-PENTANEDIOL- 1-ISOBUTYRATE		C	P	3
UNDECYLBENZENE		C	P	3
UREA, AMMONIUM SOLUTION (containing aqua ammonia)		C	SP	3
VINYL ACETATE		C	SP	3
VINYL ETHYL ETHER		C	SP	2
VINYL NEODECANOATE		C	SP	3
XYLENE		CO	P	3

List D

Pollution Category D in CH. VII/18 of BCH/IBC Codes

ALCOHOLS ETHOXYLATE	(higher secondary)	D	P	4
ALKYL (C9-C17) BENZENE MIXTURES	(straight or branched chain)	DO	P	4
AMMONIUM SULPHATE SOLUTION		D	P	4
n-AMYL ALCOHOL		D	P	4
AMYL ALCOHOL, primary		D	P	4
sec-AMYL ALCOHOL		D	P	4
BABASSU OIL		D	P	4
BEECHNUT OIL		D	P	4
BENZENETRICARBOXYLIC ACID, TRIOCTYL ESTER		D	P	4
BUTENE OLIGOMER		DO	P	4
sec-BUTYL ACETATE		D	P	4
BUTYLENE GLYCOL		D	P	4
gamma-BUTYROLACTONE		D	P	4
CALCIUM ALKYL SALICYLATE		D	P	4
CALCIUM CHLORIDE SOLUTION		D	P	4
CAPROLACTUM	(molten or aqueous solutions)	D	P	4
CHOLINE CHLORIDE SOLUTIONS		D	P	4
COCOA BUTTER OIL		D	P	4
COCONUT OIL		D	P	4
COCONUT OIL, esterefied		D	P	4
COCONUT OIL FATTY ACID		D	P	4
COCONUT OIL, FATTY ACID METHYL ESTER		D	P	4
COD LIVER OIL		D	P	4

(continued)

n-DECYL BENZENE	D	P	4
DIACETONE ALCOHOL	D	P	4
DIALKYL (C7-C9) PHTHALATES	D	P	4
DIALKYL (C9-C13) PHTHALATES	D	P	4
DIETHYLENE GLYCOL BUTYL ETHER ACETATE	D	P	4
DIETHYLENE GLYCOL DIBUTYL ETHER	D	P	4
DIETHYLENE GLYCOL ETHYL ETHER ACETATE	D	P	4
DIETHYLENE GLYCOL MONOMETHYL ETHER ACETATE	D	P	4
DIETHYLENE GLYCOL PHENOL ETHER	D	P	4
DI-(2-ETHYLHEXYL) ADIPATE	D	P	4
DI-(2-ETHYL HEXYL) PHTHALATE	D	P	4
1,4-DIHYDRO-9,10-DIHYDROXY ANTHRACENE DISODIUM SALT SOLUTION	D	P	4
DIISOBUTYL KETONE	D	P	4
DIISODECYL PHTHALATE	D	P	4
DIISONONYL ADIPATE	D	P	4
DIISOPROPYL NAPHTHALENE	DO	P	4
2,2-DIMETHYLPROPANE -1,3-DIOL	D	P	4
DIMETHYL SUCCINATE	D	P	4
DINONYL PHTHALATE	D	P	4
DIPROPYLENE GLYCOL METHYL ETHER	D	P	4
DIUNDECYL PHTHALATE	D	P	4
DODECANE	DO	P	4
2-ETHOXYETHANOL	D	P	4
ETHYL ACETATE	D	P	4
ETHYL ACETOACETATE	D	P	4
ETHYLCYCLOHEXANE	DO	P	4
ETHYLENEDIAMINE TETRAACETIC ACID, TETRASODIUM SALT, SOLUTION	D	P	4
ETHYLENE GLYCOL	D	P	4
ETHYLENE GLYCOL BUTYL ETHER ACETATE	D	P	4
ETHYLENE GLYCOL DIBUTYL ETHER	D	P	4
ETHYLENE GLYCOL ISOPROPYL ETHRE	D	P	4
ETHYLENE GLYCOL METHYL BUTYL ETHER	D	P	4
ETHYLENE GLYCOL METHYL ETHER	D	P	4
ETHYLENE GLYCOL METHYL ETHER ACETATE	D	P	4
ETHYLENE GLYCOL PHENYL ETHER	D	P	4
ETHYLENE GLYCOL PHENYL ETHER / DIETHYLENE GLYCOL PHENYL ETHER MIXTURE	D	P	4
2-ETHYLHEXANOIC ACID	D	P	4
ETHYL PROPIONATE	D	P	4
FERRIC HYDROXYETHYLETHYLENE DIAMINE TRIACETIC ACID	D	P	4
FERRIC HYDROXYETHYLETHYLENE DIAMINE TRIACETIC ACID, TRISODIUM SALT SOLN.	D	P	4
FISH OIL	D	P	4
FORMAMIDE	D	P	4
GROUND NUT OIL	D	P	4
HAZELNUT OIL	D	P	4
HEXAMETHYLENE DIAMINE ADIPATE (50% in water)	D	P	4
1-HEXANOL	D	P	4
N-(HYDROXYETHYL)ETHYLENEDIAMINE TRIACEDIC ACID, TRISODIUM SALT, SOLUTION	D	P	4
ISOAMYL ALCOHOL	D	P	4
ISOBUTYL FORMATE	D	P	4
ISOPENTANE	DO	P	4
ISOPHORONE	D	P	4
LACTIC ACID	D	P	4
LANOLIN OIL	D	P	4
3-METHOXY BUTYL ACETATE	D	P	4
METHOXYTRIGLYCOL	D	P	4
METHYL ACETOACETATE	D	P	4
METHYL tert-BUTYL ETHER	D	P	4
METHYL ETHYL KETONE	D	P	4
METHYL ISOBUTYL KETONE	D	P	4
NEATSFOOT OIL	D	P	4
NITROMETHANE	D	P	4
NONANE	DO	P	4
NONANOIC ACID	D	P	4
NONYL METHACRYLATE MONOMER	D	P	4
NUTMEG BUTTER OIL	D	P	4
OCTANE	DO	P	4
OITICICA OIL	D	P	4
OLEIC ACID	D	P	4
PALMNUT OIL	D	P	4
n-PARAFFINS (C10-C20)	DO	P	4
PEEL OIL (oranges & lemons)	D	P	4
PERILLA OIL	D	P	4
PETROLATUM	D	P	4
PILCHARD OIL	D	P	4
POLYALLKYLENE GLYCOL/POLYALKYLENE GLYCOL MONOALKYL ETHERS (mixtures)	D	P	4
POLYETHYLENE GLYCOL ALKYL ETHER	D	P	4
POLYPROPYLENE GLYCOLS	D	P	4
POPPY OIL	D	P	4
n-PROPYL ACETATE	D	P	4

(continued)

Pollution Category D in CH. VII/18 of BCH/IBC Codes *(continued)*

n-PROPYL ALCOHOL	D	P	4
PROPYLENE GLYCOL ALKYL ETHER	D	P	4
PROPYLENE GLYCOL ETHYL ETHER	D	P	4
PROPYLENE GLYCOL METHYL ETHER	D	P	4
RAISIN SEED OIL	D	P	4
SALAD OIL	D	P	4
SILICA SLURRY	D	P	4
SODIUM SILICATE SOLUTION	D	P	4
SOYABEAN OIL	D	P	4
SPERM OIL	D	P	4
TALLOW	D	P	4
TETRAETHYLENE PENTAMINE	D	P	4
TRIDECANE	D	P	4
TRIETHYLENE GLYCOL ETHYL ETHER	D	P	4
TRIISOPROPANOLAMINE	D	P	4
TRIMETHYLOL PROPANE POLYETHOXYLATE	D	P	4
TRIPROPYLENE GLYCOL METHYL ETHER	D	P	4
TRISODIUM NITRILOTRIACETATE SOLUTION	D	P	4
UREA, AMMONIUM NITRATE SOLUTION	D	P	4
UREA, AMMONIUM PHOSPHATE SOLUTION	D	P	4
WALNUT OIL	D	P	4
WAXES	D	P	4
WHALE OIL	D	P	4

List D1

Pollution Category D in CH. VI/17 of BCH/IBC Codes

ACRYLAMIDE SOLUTION	(50% or less)	D	S	2
ACRYLIC ACID		D	S	3
ADIPONITRILE		D	S	3
2-(2-AMINOETHOXY) ETHANOL		D	S	3
AMINOETHYL ETHANOLAMINE		D	S	3
N-AMINOETHYL PIPERAZINE		D	S	3
2-AMINO-2-METHYL-1-PROPANOL	(90% or less)	D	S	3
AMMONIUM NITRATE SOLUTION	(93% or less)	D	S	2
BENZENESULPHONYL CHLORIDE		D	S	3
n-BUTYL ACRYLATE		D	S	3
BUTYL/DECYL/CETYL-EICOSYL METHACRYLATE MIXTURE		D	S	3
BUTYL METHACRYLATE		D	S	3
CASHEW NUT SHELL OIL (untreated)		D	S	3
CHLOROHYDRINS (crude)		D	S	2
COAL TAR PITCH, (molten)		D	S	3
CYCLOHEXANONE		D	S	3
DICHLOROMETHANE		D	S	3
2,2-DICHLOROPROPIONIC ACID		D	S	3
DIETHYLENETRIAMINE		D	S	3
N,N-DIMETHYLACETAMIDE SOLUTION (40% or less)		D	S	3
DIMETHYLETHANOLAMINE		D	S	3
DIMETHYLFORMAMIDE		D	S	3
1,4-DIOXANE		D	S	2
ETHANOLAMINE		D	S	3
N-ETHYLCYCLOHEXYLAMINE		D	S	3
ETHYLENE CYANOHYDRIN		D	S	3
ETHYLENE OXIDE/PROPYLENE OXIDE MIX. (ETHYLENE OXIDE CONTENT NOT > 30%)		D	S	2
2-ETHYLHEXYL ACRYLATE		D	S	3
ETHYL METHACRYLATE		D	S	3
FORMIC ACID		D	S	3
GLUTARALDEHYDE SOLUTIONS	(50% or less)	D	S	3
HYDROCHLORIC ACID		D	S	3
ISOBUTYL ACRYLATE		D	S	2
ISOPHORONE DIAMINE		D	S	3
ISOPROPYL ETHER		D	S	3
MALEIC ANHYDRIDE		D	S	3
MESITYL OXIDE		D	S	3
METHACRYLIC ACID		D	S	3
METHYL FORMATE		D	S	2
METHYL METHACRYLATE		D	S	2
MORPHOLINE		D	S	3
1-or 2-NITROPROPANE		D	S	3
NITROPROPANE (60%)/ NITROETHANE (40%) MIXTURES		D	S	3
PHOSPHORIC ACID		D	S	3
POLYMETHYLENE POLYPHENYL ISOCYANATE		D	S	2
beta-PROPIOLACTONE		D	S	3
PROPIONALDEHYDE		D	S	3
PROPIONIC ACID		D	S	3

(continued)

Pollution Category D1 in CH. VI/17 of BCH/IBC Codes *(continued)*

PROPYLENE OXIDE	D	S	2
SODIUM HYDROGEN SULPHITE SOLUTION (35% or less)	D	S	3
SODIUM HYDROXIDE SOLUTION	D	S	3
TETRAHYDROFURAN	D	S	3
TRIETHANOLAMINE	D	S	3
TRIETHYLENETETRAMINE	D	S	3
TRIETHYL PHOSPHITE	D	S	3
TRIMETHYLACETIC ACID	D	S	3
TRIMETHYLHEXAMETHYLENE DIAMINE (2,2,4-and-2,4,4- ISOMERS)	D	S	3
n-VALERALDEHYDE	D	S	3

List E

Pollution Category E Non-pollutants in CH. VII/18 of BCH/IBC Codes

ACETONE	III	P	4
ALCOHOLIC BEVERAGES	III	P	4
ALCOHOLS, C13 & above as individuals & mixtures	III	P	4
ALCOHOL ETHOXYLATE (C6-C17 secondary)	III	P	4
AMMONIUM HYDROGEN PHOSPHATE	III	P	4
tert-AMYL ALCOHOL	III	P	4
APPLE JUICE	III	P	4
BEHENYL ALCOHOL	III	P	4
n-BUTYL ALCOHOL	III	P	4
sec-BUTYL ALCOHOL	III	P	4
tert-BUTYL ALCOHOL	III	P	4
BUTYL METHYL KETONE	III	P	4
BUTYL STEARATE	III	P	4
CALCIUM BROMIDE SOLUTION	III	P	4
CALCIUM CARBONATE SLURRY	III	P	4
CETYL STEARYL ALCOHOL	III	P	4
CHLORONATED PARAFFINS (C14-C17) WITH 52% CHLORINE	III	P	4
DEXTROSE SOLUTION	III	P	4
DICYCLOPENTADIENE	III	P	4
DIETHYLENE GLYCOL	III	P	4
DIETHYLENE GLYCOL BUTYL ETHER	III	P	4
DIETHYLENE GLYCOL DIETHYL ETHER	III	P	4
DIETHYLENE GLYCOL ETHYL ETHER	III	P	4
DIETHYLENETRIAMINE PENTOACETIC ACID, PENTASODIUM SALT 40% SOLUTION	III	P	4
DI-n-HEXYL ADIPATE	III	P	4
DIHEPTYL PHTHALATE	III	P	4
DIHEXYL PHTHALATE	III	P	4
DIISOOCTYL PHTHALATE	III	P	4
DIOCTYL PHTHALATE	III	P	4
DIPROPYLENE GLYCOL	III	P	4
ETHYL ALCOHOL	III	P	4
ETHYLENE CARBONATE	III	P	4
ETHYLENE GLYCOL BUTYL ETHER	III	P	4
ETHYLENE GLYCOL ETHYL ETHER (select 2-ETHOXYETHANOL)	III	P	4
ETHYLENE GLYCOL tert BUTYL ETHER	III	P	4
ETHYLENE PROPYLENE COPOLYMER (in liquid mixture)	III	P	4
ETHYLENE-VINYLACETATE COPOLYMER (emulsion)	III	P	4
GLYCERIN	III	P	4
GLYCEROL POLYALKOXYLATE	III	P	4
GLYCEROL TRIACETATE	III	P	4
GLYCINE, SODIUM SALT, SOLUTION	III	P	4
GLYCOL DIACETATE	III	P	4
GLYCOL TRIACETATE	III	P	4
iso-HEPTANE	III	P	4
n-HEPTANE	III	P	4
HEXAMETHYLENE GLYCOL	III	P	4
n-HEXANE	III	P	4
1,6-HEXANEDIOL	III	P	4
HEXYLENE GLYCOL	III	P	4
ISOBUTYL ALCOHOL	III	P	4
ISOPROPYL ACETATE	III	P	4
ISOPROPYL ALCOHOL	III	P	4
KAOLIN CLAY SLURRY	III	P	4
KAOLIN SLURRY	III	P	4
LARD	III	P	4
LATEX (carboxylated styrene- butadiene copolymer)	III	P	4
LIGNIN SULPHONIC ACID, SALT (low COD) SOLUTION	III	P	4
MAGNESIUM CHLORIDE SOLUTION	III	P	4
MAGNESIUM HYDROXIDE SLURRY	III	P	4
3-METHOXY-1-BUTANOL	III	P	4

(continued)

Pollution Category E Non-pollutants in CH. VII/18 of BCH/IBC Codes *(continued)*

METHYL ACETATE	III	P	4
METHYL ALCOHOL	III	P	4
3-METHYL-3-METHOXY BUTANOL	III	P	4
3-METHYL-3-METHOXY BUTYL ACETATE	III	P	4
MOLASSES	III	P	4
OLEFINS (C13 and above, all isomers)	III	P	4
PARAFFIN WAX	III	P	4
POLYALUMINIUM CHLORIDE SOLUTION	III	P	4
POLYBUTENE	III	P	4
POLYETHYLENE GLYCOL	III	P	4
POLYETHYLENE GLYCOL DIMETHYL ETHER	III	P	4
POLYISOBUTYLENE	III	P	4
POLYPROPYLENE	III	P	4
POLYPROPYLENE GLYCOLS METHYL ETHER	III	P	4
POLYSILOXANE	III	P	4
POTASSIUM CHLORIDE	III	P	4
PROPYLENE BUTYLENE COPOLYMER	III	P	4
1,2-PROPYLENE GLYCOL	III	P	4
PROPYLENE TETRAMER	III	P	4
SODIUM ALUMINOSILICATE SLURRY	III	P	4
SORBITOL SOLUTION	III	P	4
SULPHOLANE	III	P	4
TETRAETHYLENE GLYCOL	III	P	4
TRIDECANOIC	III	P	4
TRIDECANOL	III	P	4
TRIETHYLENE GLYCOL	III	P	4
TRIETHYLENE GLYCOL BUTYL ETHER	III	P	4
TRIPROPYLENE GLYCOL	III	P	4
UREA-FORMALDEHYDE RESIN SOLUTION	III	P	4
UREA RESIN SOLUTION	III	P	4
VEGETABLE PROTEIN HYDROLIZED SOLUTION	III	P	4
WATER	III	P	4
WINES	III	P	4

List E1

Pollution Category E Non-Pollutants in CH. VI/17 of BCH/IBC Codes

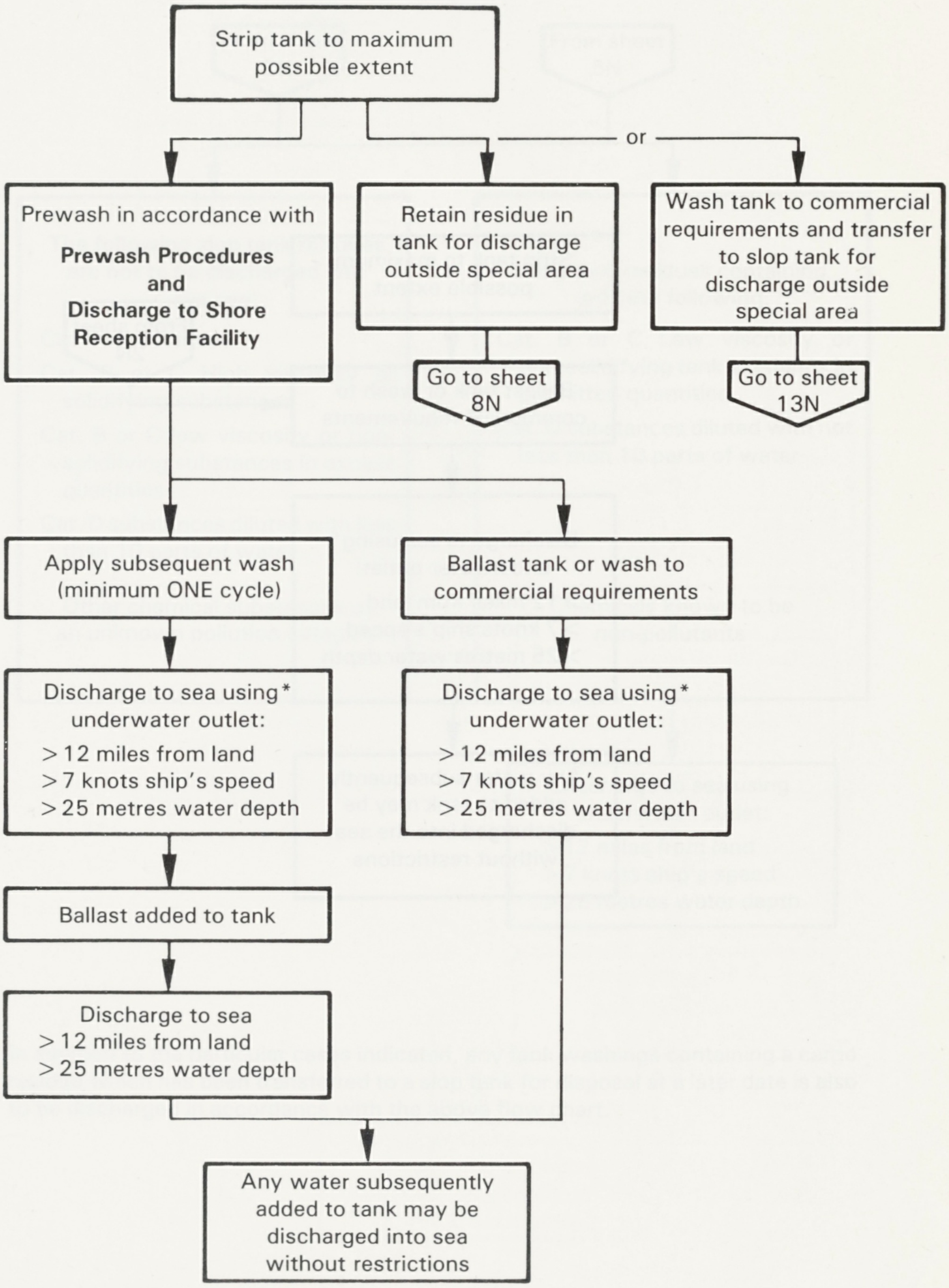
ACETONITRILE	III	S	2
CETYL/EICOSYL METHACRYLATE MIXTURE	III	S	3
DIETHANOLAMINE	III	S	3
DIETHYL ETHER	III	S	2
DODECYL METHACRYLATE	III	S	3
DODECYL/PENTADECYL METHACRYLATE MIXTURE	III	S	3
2-METHYL-2-HYDROXY-3-BUTYNE	III	S	3
SODIUM CHLORATE SOLUTIONS (50% or less)	III	S	3
SULPHUR, (molten)	III	S	3

APPENDIX B

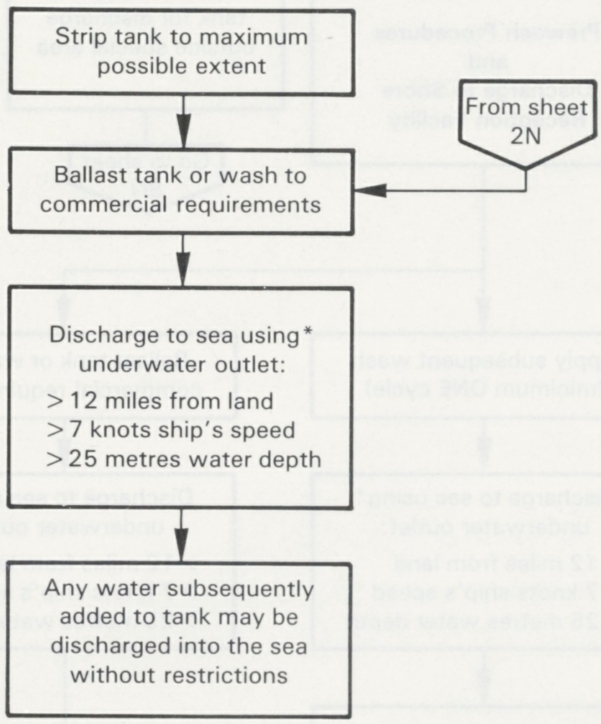
FLOW CHARTS

Flow Chart Sheet No.	
2N 8N 13N	Extracts from ICD/ICL 182 Guidance Document for NEW SHIPS
4E 23E 24E 25E 26E	Extracts from ICD/ICL 182 Guidance Document for EXISTING SHIPS. These charts are applicable to ships operating on the higher stripping residue quantity permissible until 2nd October 1994.

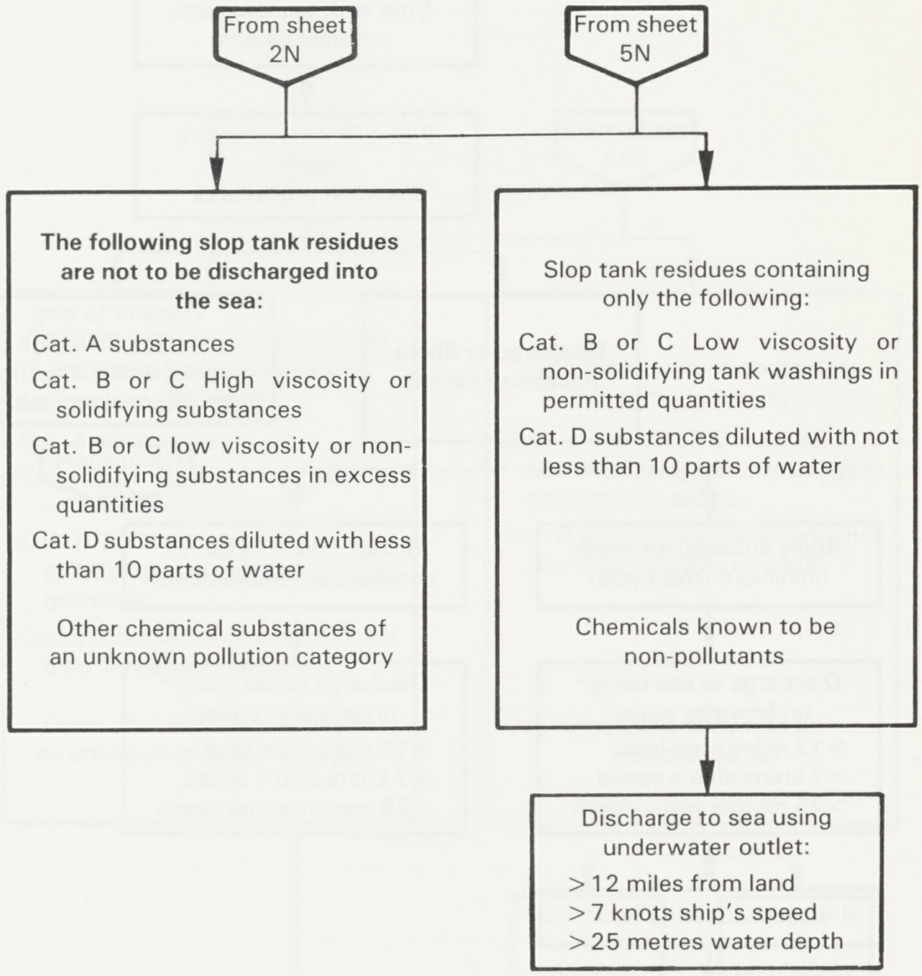
Cargo Tank Residue Disposal Procedure Flow Chart—Sheet 2N	Inside Special Area	
Low Viscosity: Less than 25mPa.S at the unloading temperature **Non-solidifying:** Melting point $< 15°C$ and unloaded at $> 5°C$ above melting point Melting point $\geqslant 15°C$ and unloaded at $> 10°C$ above melting point	**Category** Low Viscosity or Non-solidifying	**B**
	Design Stripping Quantity (m^3)	**0.1**



Cargo Tank Residue Disposal Procedure Flow Chart—Sheet 8N		Outside Special Area	
Low Viscosity:	Less than 25mPa.S at the unloading temperature	Category Low Viscosity or Non-solidifying	B
Non-solidifying:	Melting point $<15\,^{\circ}C$ and unloaded at $>5\,^{\circ}C$ above melting point Melting point $\geqslant 15\,^{\circ}C$ and unloaded at $>10\,^{\circ}C$ above melting point	Design Stripping Quantity (m^3)	0.1

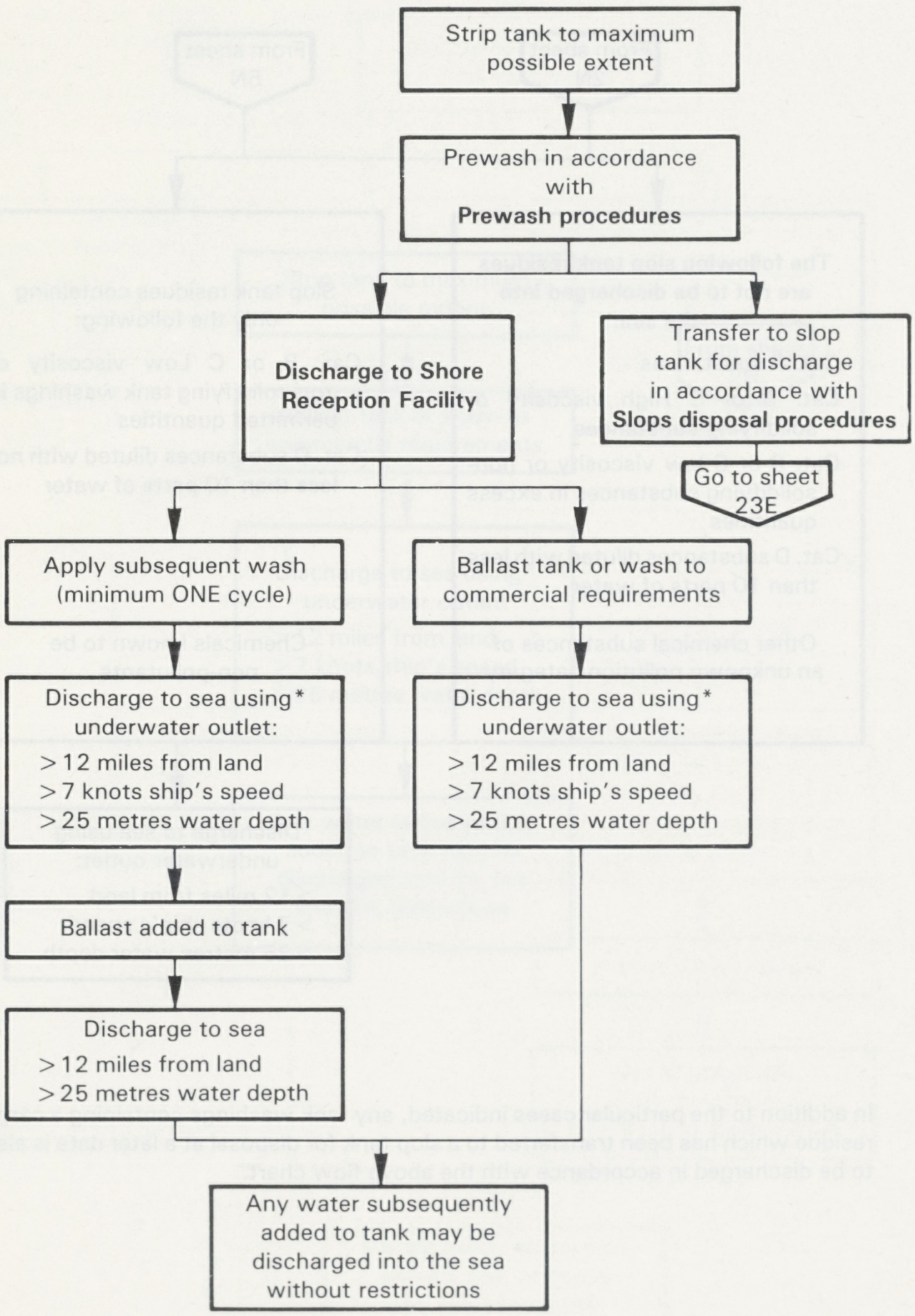


Slop Tank Residue Disposal Procedure Flow Chart—Sheet 13N	Outside Special Area

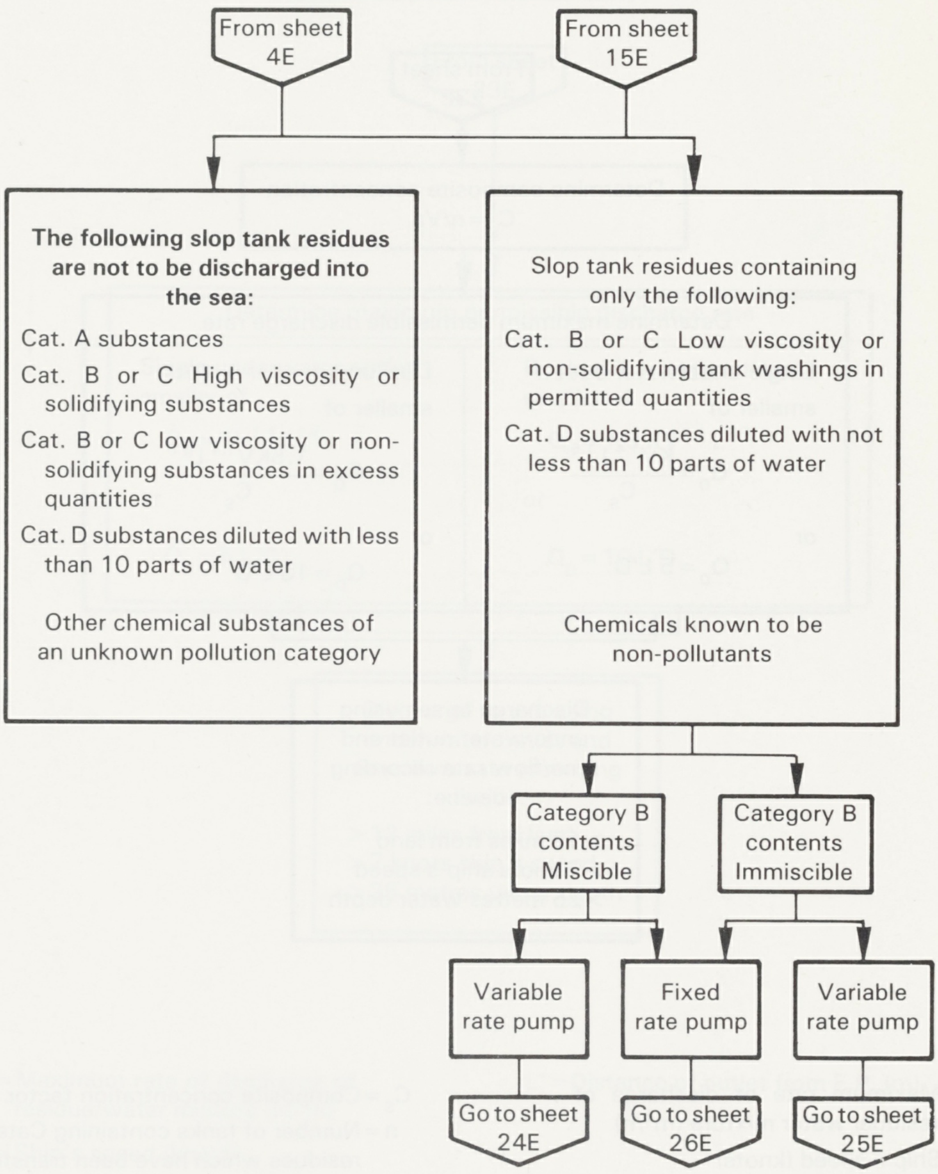


In addition to the particular cases indicated, any tank washings containing a cargo residue which has been transferred to a slop tank for disposal at a later date is also to be discharged in accordance with the above flow chart.

Cargo Tank Residue Disposal Procedure Flow Chart—Sheet 4E	Inside Special Area (Permissible until 2/10/94)
Low Viscosity: Less than 25mPa.S at the unloading temperature **Non-solidifying:** Melting point $<15^\circ C$ and unloaded at $>5^\circ C$ above melting point Melting point $\geqslant 15^\circ C$ and unloaded at $>10^\circ C$ above melting point	**Category** Low Viscosity or Non-solidifying — **B** **Design Stripping Quantity (m^3)** — **1.0**



Slop Tank Residue Disposal Procedure Flow Chart—Sheet 23E	Outside Special Area (Permissible until 2/10/94)	
	Category Low Viscosity or Non-solidifying	**B**
	Design Stripping Quantity (m^3)	**1.0**



Slop Tank Residue Disposal Procedure Flow Chart – Sheet 24E		Outside Special Area (Permissible until 2/10/94)
	Category Low Viscosity or Non-solidifying	B
	Design Stripping Quantity (m^3)	1.0

Slop Tank Equipped with Variable Capacity Discharge Pump

MISCIBLE water/residue mixtures

From sheet 23E

Determine composite concentration
$C_s = n/Vr$

Determine maximum permissible discharge rate

Single underwater outlet: smaller of

$$Q_D = \frac{KV^{1.4}\,L^{1.6}}{C_S}$$

or

$$Q_D = 5\,L'D$$

Dual underwater outlets: smaller of

$$Q_D = \frac{1.5KV^{1.4}\,L^{1.6}}{C_S}$$

or

$$Q_D = 10\,L'D$$

Discharge to sea using underwater outlet and time/flow rate recording device:
- >12 miles from land
- >7 knots ship's speed
- >25 metres water depth

Q_D = Maximum rate of discharge of residue/ water mixture (m^3/h)

V = Ship's speed (knots)

L = Ship's length (m)

$K = 4.3 \times 10^{-5}$

D = Calculated diameter of underwater outlet (m)

C_S = Composite concentration factor.

n = Number of tanks containing Category B residues which have been transferred to slop tank

V_r = Volume of residue/water mixtures in the slop tank prior to discharge (m^3)

L' = Distance of outlet from F.P. (m)

Slop Tank Residue Disposal Procedure Flow Chart—Sheet 25E	Outside Special Area (Permissible until 2/10/94)	
	Category Low Viscosity or Non-solidifying	B
	Design Stripping Quantity (m^3)	1.0

Slop Tank Equipped with Variable Capacity Discharge Pump

IMMISCIBLE
water/residue mixtures

From sheet 23E

Determine maximum permissible discharge rate

Single underwater outlet: smaller of	Dual underwater outlets: smaller of
$Q_D = KV^{1.4} L^{1.6}$	$Q_D = 1.5\, KV^{1.4} L^{1.6}$
or	or
$Q_D = 5\, L'D$	$Q_D = 10\, L'D$

Discharge to sea using underwater outlet and time/flow rate recording device:

- $>$12 miles from land
- $>$7 knots ship's speed
- $>$25 metres water depth

Q_D = Maximum rate of discharge of residue/water mixture (m^3/h)

V = Ship's speed (knots)

L = Ship's length (m)

$K = 4.3 \times 10^{-5}$

D = Calculated diameter of underwater outlet (m)

L' = Distance of outlet from F.P. (m)

Slop Tank Residue Disposal Procedure Flow Chart – Sheet 26E	Outside Special Area (Permissible until 2/10/94)	
	Category Low Viscosity or Non-solidifying	B
	Design Stripping Quantity (m^3)	1.0

Slop Tank Equipped with Constant Capacity (fixed rate) Discharge Pump

MISCIBLE or IMMISCIBLE
water/residue mixtures

From sheet 23E

Discharge to sea using underwater outlet and time recording device:

- $>$ 12 miles from land
- $>$ 7 knots ship's speed
- $>$ 25 metres water depth

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Lloyd's Register Technical Association

STRESS ANALYSIS OF HEATED SUBMARINE PIPELINES

D. M. Richards

Paper No. 4. Session 1986–87

FOR PRIVATE CIRCULATION AMONGST THE STAFF ONLY

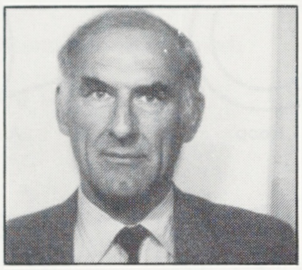
Written contributions to the discussion of this paper are invited from members of the Lloyd's Register Technical Association. To ensure inclusion in the discussion paper, the contributions should be received by the Hon. Secretary in London not later than the 30th April, 1987.

Hon. Sec. C. M. Magill
71 Fenchurch Street, London, EC3M 4BS

STRESS ANALYSIS OF HEATED SUBMARINE PIPELINES

by

D. M. Richards



D. M. Richards studied aeronautical engineering at City University, London, and Cranfield Institute of Technology. After serving an engineering apprenticeship at Handley Page Ltd. and a short term research appointment at the Massachusetts Institute of Technology, he was appointed Research Fellow and subsequently Lecturer in Aircraft Design at Cranfield.

He joined the Society as a specialist structural engineer in 1977, and has since been responsible for the development of analytical procedures for pipeline design appraisal and other related problems, including shell buckling.

TABLE OF CONTENTS

1. INTRODUCTION

The exploitation of offshore hydrocarbon resources has required the design and installation of extensive networks of submarine pipelines for the onshore delivery of oil and gas.

Located on the seabed, often in deep waters, these pipelines require complex and expensive laying operations, and their comparative inaccessibility makes routine inspection difficult.

Thus accurate and economical design is of great importance, since mistakes may be difficult to identify, and expensive to rectify.

The codes by which submarine pipeline design is regulated are derived directly from land-based codes, with additions mostly concerned with hydrodynamic loading effects. There have been few significant advances in stress analysis procedures embodied in the revised codes.

The objective of this paper is to review the principles and properties involved in pipeline mechanical behaviour under load and to illustrate the approach to design appraisal adopted in the Ocean Engineering Department by discussing solutions to certain problems concerned specifically with high temperature effects on pipeline strength.

The economic development of small oilfields in particular may not justify the installation of local processing facilities. Thus oil may require to be exported hot by small diameter, often insulated pipelines to distant locations.

The application of the simplistic criteria embodied in many codes in this situation can lead to unrealistic designs whose assumed mechanical behaviour bears little relation to established principles of solid mechanics.

The following examples are intended to give some indication of ways in which these often rather intractable problems may be solved, and the results applied to the development of rational design procedures.

2. IDEALISATION

2.1 Forces

The elementary quantities involved in the analysis of pipelines differ little from those required to describe the behaviour of other forms of construction.

A pipeline may be regarded for almost all analytical purposes as a slender, pressurised, thin walled tube. As such, the process of describing mathematically forces, stresses and deformations involves fewer compromises than is usually the case in stress analysis.

The central purpose requiring the transport of pressurised fluids implies that internal pressure should be regarded as a primary loading action and yielding under hoop stress as the failure mode of greatest concern. However, since the pipeline will be laid typically in an irregular path on an uneven seabed subject to ocean currents, and the transported fluid may be as much as 90°C above the ambient temperature, other significant loading actions will arise and must be considered in any assessment of pipeline strength.

Any deviation from a straight line will give rise to bending and shear forces, which will be further exaggerated by the axial pressure force arising from the contained fluid and axial wall forces.

Submarine pipelines are designed typically to be negatively buoyant by the addition of a reinforced concrete weight coat if necessary. This will increase bending forces where free spans occur between points of contact with the seabed.

Thermal expansion or hydrodynamic forces will tend to move the pipeline in the horizontal plane. This movement will be resisted by soil friction and the associated non-linear behaviour gives rise to some of the most difficult problems in pipeline analysis.

Contact with the seabed itself is also a source of analytical difficulty, since the existence and extent of contact are frequently unknown *a priori*, and must form part of the solution.

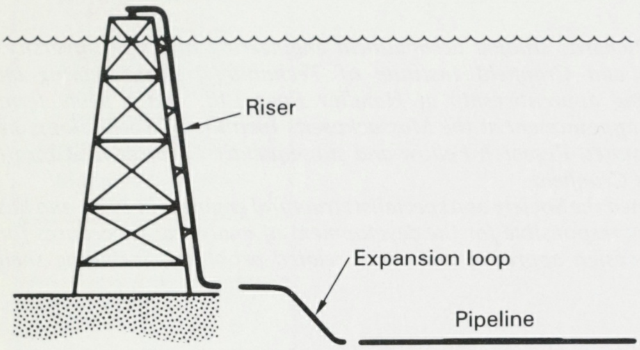


Fig. 1 Submarine Pipeline Components

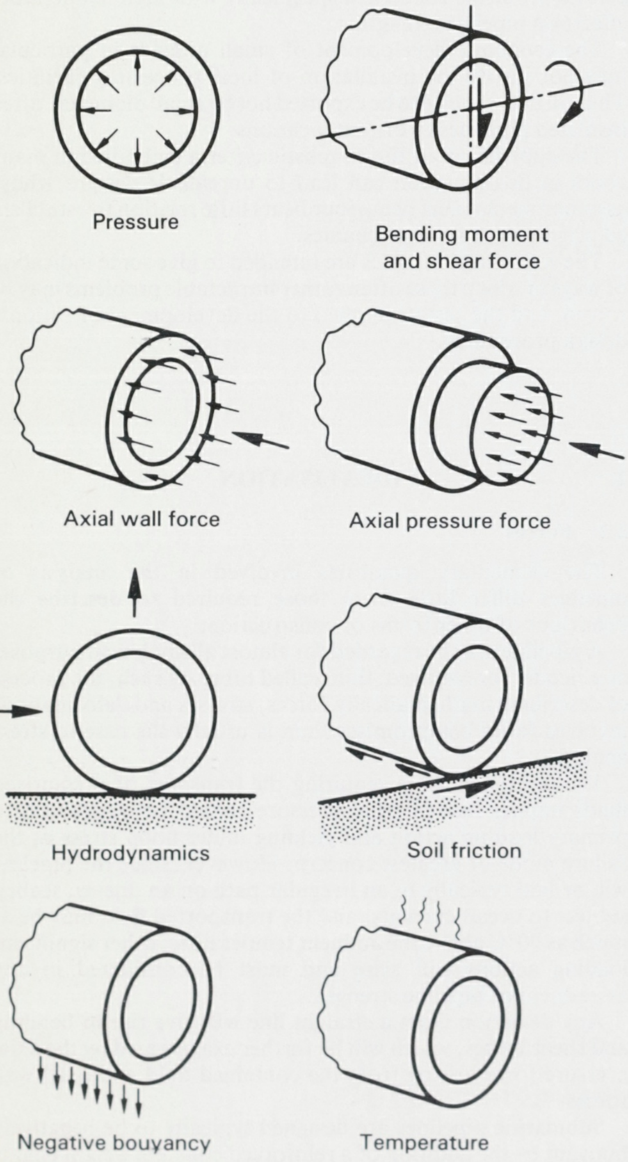


Fig. 2 Pipeline loading Actions

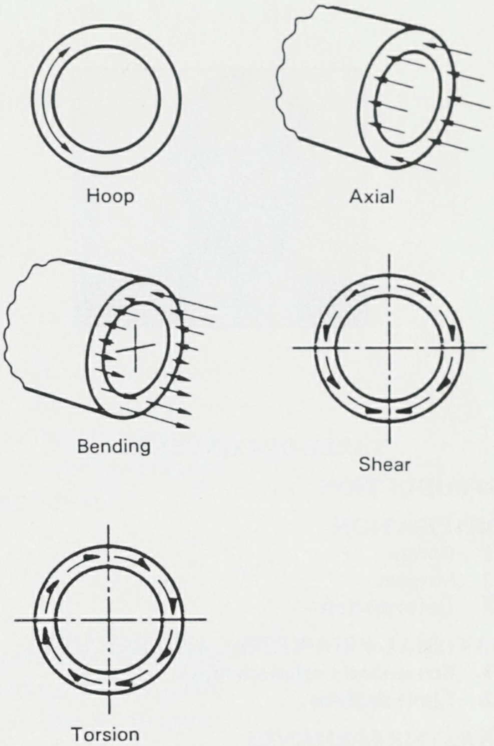


Fig. 3 Pipeline Wall Stresses

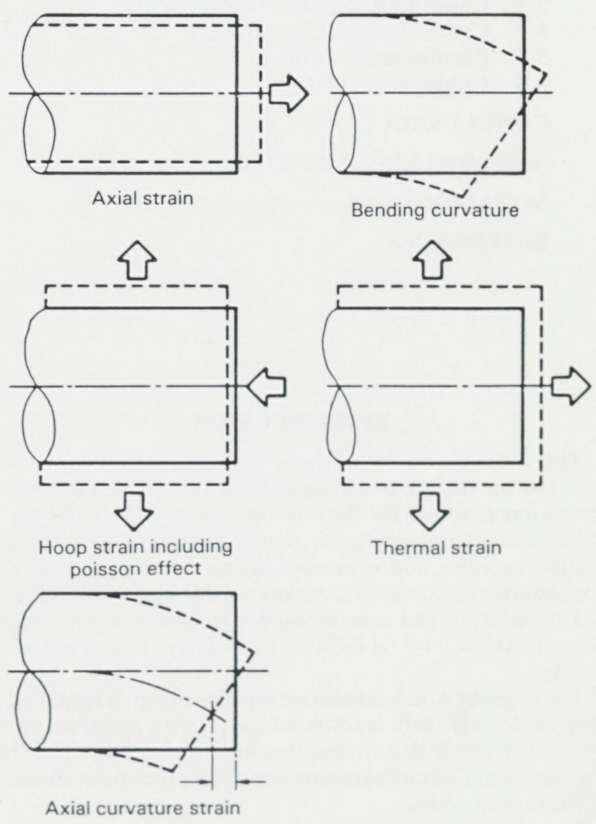


Fig. 4 Pipeline Deformations

2.2 Stresses

Pipeline diameter to wall thickness ratios are typically within the range 20–30. As such, thin walled tube theory may be applied with acceptable accuracy. On this basis, the significant stresses are hoop stresses, axial direct stress, and tangential shear stress.

Although shear stress arising from transverse shear forces and torsion are rarely important, calculation is simple and should be included in any stressing routine for completeness.

The following relationships may be used:

$$\sigma_a = -\frac{P_w}{A} \pm \frac{MD_o}{2I} \tag{1}$$

$$\sigma_h = \frac{pD_i}{D_o - D_i} \tag{2}$$

$$\tau = \frac{T_o}{J} \pm \frac{\pi S}{D_i(D_o - D_i)} \tag{3}$$

$$A = \frac{\pi}{4}(D_o^{\,2} - D_i^{\,2}) \tag{4}$$

$$I = \frac{\pi}{64}(D_o^{\,4} - D_i^{\,4}) \tag{5}$$

$$J = \frac{\pi}{32}(D_o - D_i)(D_o + D_i)^2 \tag{6}$$

(Please refer to Notation listing for details)

In order to avoid errors arising from any assumption about force combination, O.E.D. pipeline stress analysis routines normally scan the pipe periphery at 5° intervals to detect maximum stress combinations.

2.3 Deformations

An integral part of stress analysis is the satisfaction of compatibility of displacement between components and with boundary restraints. This aspect is particularly important for pipelines, due to the contact and frictional phenomena previously discussed.

Apart from the axial strain arising from axial wall forces, and curvature due to bending, additional axial strains from the following sources should be included:

(i) Axial strain due to hoop stress:

$$\epsilon_p = -\nu\frac{\sigma_h}{E} \tag{7}$$

(ii) Thermal expansion:

$$\epsilon_t = \alpha T \tag{8}$$

(iii) Axial curvature strain, which arises from the shortening associated with bending deformations, given by:

$$\epsilon_c = -\frac{1}{2}\int\left(\frac{dy}{dx}\right)^2 dx \tag{9}$$

3. MATERIAL PROPERTIES AND FAILURE

3.1 Stress-strain relations

Most design codes require the restriction of working stresses to certain given proportions of yield stress.

Elastic behaviour is assumed up to yield, and strain hardening effects on stress analysis are not contemplated. However, the API 5LX specification defines yield stress (σ_y) as the 0.5% proof stress, i.e. the stress at which a plastic strain of 0.005 occurs. Also, the ratio of yield to ultimate is restricted to a maximum of 0.85, and a minimum elongation at failure of the order of 20% is required.

Clearly, significant plasticity could occur before yield is reached for a minimum specification material, which could be important for a detailed analysis of stress distribution or buckling behaviour.

A consistent approach to elastic/plastic analysis may be established by utilising the Ramberg-Osgood stress-strain relationship, which takes the form:

$$\epsilon = \frac{\sigma}{E} + a\left(\frac{\sigma}{\sigma_y}\right)^n \tag{10}$$

The API pipeline steel requirements are satisfied if we set $a = 0.005$ and $n = 22.7$ in (10) giving the stress-strain property illustrated in Fig. 5.

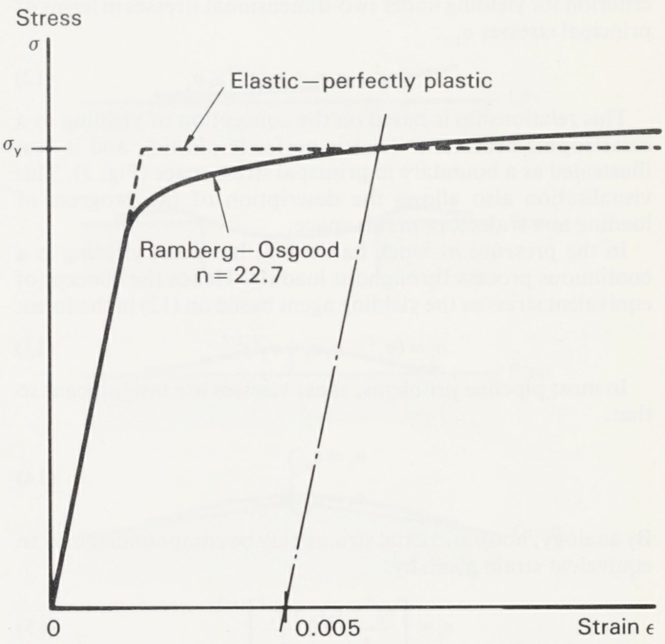


Fig. 5 Idealised Pipeline Steel Stress—Strain Behaviour

Multi-axial stress-strain behaviour may be characterised by generalising (10) by introducing the concepts of generalised stress and strain σ_e and ϵ_e which are functions of principal stress. Thus from (10):

$$\epsilon_e = \frac{\sigma_e}{E} + a\left(\frac{\sigma_e}{\sigma_y}\right)^n \tag{11}$$

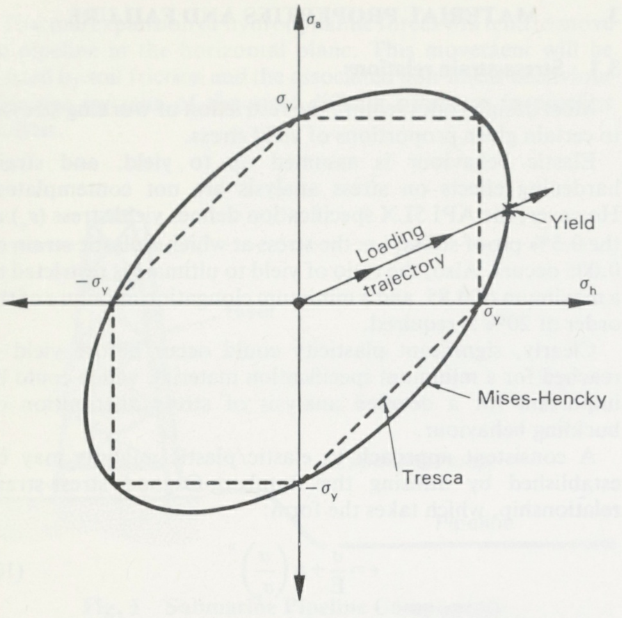


Fig. 6 Yield Theories for Biaxial Stress

The most widely accepted theory of yielding under multi-axial stresses is Mises-Hencky which gives the following criterion for yielding under two-dimensional stresses in terms of principal stresses $\sigma_{1,2}$:

$$(\sigma_1{}^2 - \sigma_1\sigma_2 + \sigma_2{}^2)^{1/2} \leqslant \sigma_y \tag{12}$$

This relationship is based on the conception of yielding as a catastrophic occurrence (elastic-perfectly plastic), and is best illustrated as a boundary in principal stress space (Fig. 2). This visualisation also allows the description of the progress of loading as a trajectory in this space.

In the presence of work hardening, however, yielding is a continuous process throughout loading. Hence the concept of equivalent stress as the yielding agent based on (12) in the form:

$$\sigma_e = (\sigma_1{}^2 - \sigma_1\sigma_2 + \sigma_2{}^2)^{1/2} \tag{13}$$

In most pipeline problems, shear stresses are insignificant so that:

$$\left.\begin{aligned} \sigma_1 &= \sigma_h \\ \sigma_2 &= \sigma_a \end{aligned}\right\} \tag{14}$$

By analogy, hoop and axial strains may be compounded into an equivalent strain given by:

$$\epsilon_e = \left[\frac{\epsilon_h{}^2 - \epsilon_h\epsilon_a + \epsilon_a{}^2}{1 + \nu + \nu^2}\right]^{1/2} \tag{15}$$

Equations (13) to (15) may be used together to provide a basis for the analysis of elasto-plastic behaviour in pipelines which is consistent with uniaxial behaviour, code criteria, and established yield theories.

3.2 Limit analysis

A number of different modes of failure must be considered in pipeline design. All can be classified in terms of two ultimate effects—wall fracture or loss of cross-section shape, which could be described alternatively as overstressing or buckling.

In practice, both phenomena are likely to be present in some measure in a damaged pipeline.

The concept of limit analysis, concerned with ultimate rather than intermediate states is useful in considering critical combinations of load. For this purpose the simplification of elastic-perfectly plastic yield behaviour is adequate.

When lateral wall pressure alone is acting, failure will occur when hoop stress reaches yield, from (2):

$$p^* = \pm\frac{(D_o - D_i)}{D_i} \cdot \sigma_y \tag{16}$$

Since the complete stress system is required for equilibrium this relationship represents a true limit state.

The same is true for the corresponding axial load condition:

$$P^* = \pm\frac{\pi}{4} D_o{}^2 - D_i{}^2)\,\sigma_y \tag{17}$$

When the maximum stress due to bending moment reaches yield, however, the pipe section has a reserve of strength available from the redistribution of axial stresses. Thus two conditions can be defined, the elastic yield moment:

$$M_e^* = \pm\frac{\pi}{2}\frac{(D_o{}^4 - D_i{}^4)}{D_o}\,\sigma_y \tag{18}$$

and the limit moment

$$M^* = \pm\frac{1}{8}(D_o - D_i)(D_o + D_i)^2\,\sigma_y \tag{19}$$

For tubes of the usual pipeline proportions M* will be about 32% greater than M_e^*, so that clearly to regard the elastic yield moment as failure is highly conservative.

The presence of an internal pressure will make significant differences to strength under axial and bending forces, since the individual stress components will interact according to the Mises-Hencky relationship. Thus with a pressure p acting, axial load will be limited by:

$$\frac{1}{2}\left\{\frac{p}{p^*} - \left[4 - 3\left(\frac{p}{p^*}\right)^2\right]^{1/2}\right\} < \frac{P}{P^*} < \left\{\frac{p}{p^*} + \left[4 - 3\left(\frac{p}{p^*}\right)^2\right]^{1/2}\right\} \tag{20}$$

Under a bending moment, tensile and compressive parts of the cross-section will be subject to different yield stresses, given by:

$$\left.\begin{aligned} r_t &= \frac{\sigma_{yt}}{\sigma_y} = \frac{1}{2}\left\{\frac{p}{p^*} + \left[4 - 3\left(\frac{p}{p^*}\right)^2\right]^{1/2}\right\} \\ r_c &= \frac{\sigma_{yc}}{\sigma_y} = \frac{1}{2}\left\{\frac{p}{p^*} - \left[4 - 3\left(\frac{p}{p^*}\right)^2\right]^{1/2}\right\} \end{aligned}\right\} \tag{21}$$

and ultimate moment will be given by

$$\frac{M}{M^*} = \frac{1}{2}(r_c + r_t)\cos\theta \tag{22}$$

where

$$\theta = -\frac{\pi\left[\dfrac{P}{P^*} + \dfrac{1}{2}(r_c - r_t)\right]}{(r_c + r_t)} \tag{23}$$

Since the Mises-Hencky relationship implies in some cases that different loading actions can combine to give increased strength, further constraints must be included to allow for uncertainties in the relationship between forces. This is covered by the conditions:

$$\left.\begin{aligned} -1 &< \frac{p}{p^*} < 1 \\ -1 &< \frac{P}{P^*} < 1 \\ -1 &< \frac{M}{M^*} < 1 \end{aligned}\right\} \tag{24}$$

The resulting composite limit state for all three loading actions is illustrated in Fig. 7.

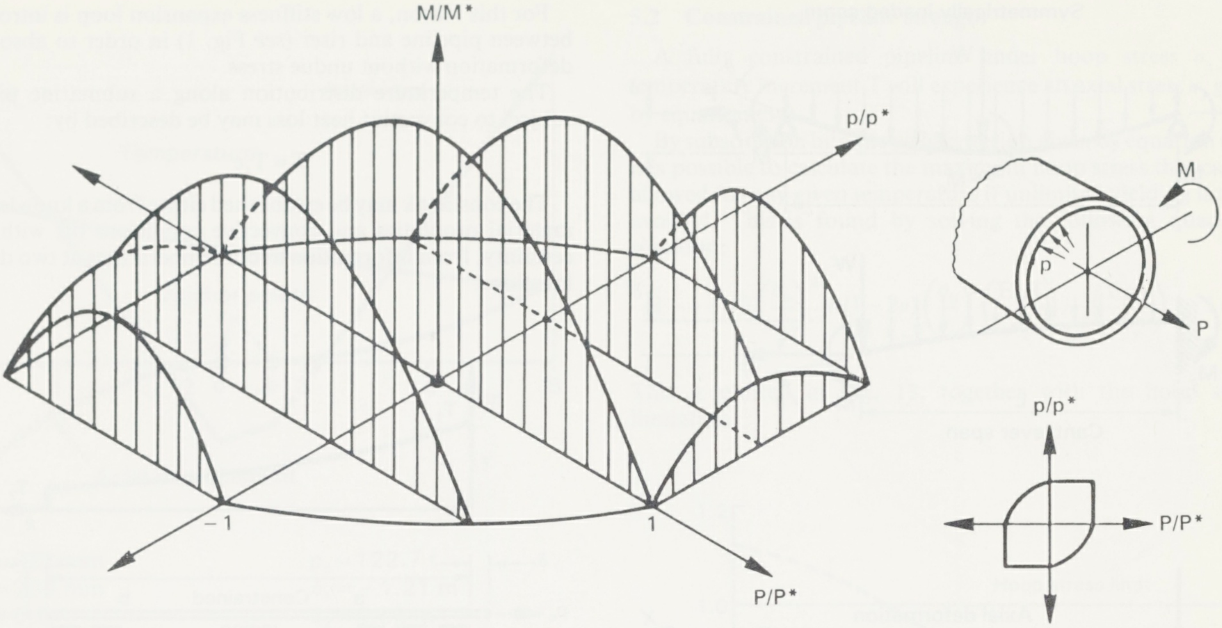


Fig. 7 Limit State Boundaries for Pressurised Pipeline

4. ANALYSIS ELEMENTS

4.1 Bending

Many stress analysis problems arise from the pipeline interaction with seabed topography (Fig. 8).

Slender beam-column small displacement theory may be used, for which deformations are based on the fundamental beam equilibrium equation

$$EI\frac{d^4y}{dx^4} - P\frac{d^2y}{dx^2} - w = 0 \tag{25}$$

Two particular solutions to (25) have been found to be applicable to a wide range of pipeline problems.

For the symmetrically loaded simply supported beam shown in Fig. 9, boundary deformations and mid-span bending moment are conveniently expressed in the following form:

$$EI\theta = \phi_1 M\left(\frac{L}{j}\right) + \phi_2 W\left(\frac{L}{j}\right)^2 + \phi_3 w\left(\frac{L}{j}\right)^3 \tag{26}$$

$$EI\delta = \phi_4 M\left(\frac{L}{j}\right)^2 + \phi_5 W\left(\frac{L}{j}\right)^3 + \phi_6 w\left(\frac{L}{j}\right)^4 \tag{27}$$

$$M_o = \phi_7 M + \phi_8 W\left(\frac{L}{j}\right) + \phi_9 w\left(\frac{L}{j}\right)^2 \tag{28}$$

where

$$j = L\left[\frac{P}{EI}\right]^{1/2} \tag{29}$$

Likewise, for the cantilever span:

$$EI\theta = \psi_1 M\left(\frac{L}{j}\right) + \psi_2 W\left(\frac{L}{j}\right)^2 + \psi_3 w\left(\frac{L}{j}\right)^3 \tag{30}$$

$$EI\delta = \psi_4 M\left(\frac{L}{j}\right)^2 + \psi_5 W\left(\frac{L}{j}\right)^3 + \psi_6 w\left(\frac{L}{j}\right)^4 \tag{31}$$

$$M_o = \psi_7 M + \psi_8 W\left(\frac{L}{j}\right) + \psi_9 w\left(\frac{L}{j}\right)^2 \tag{32}$$

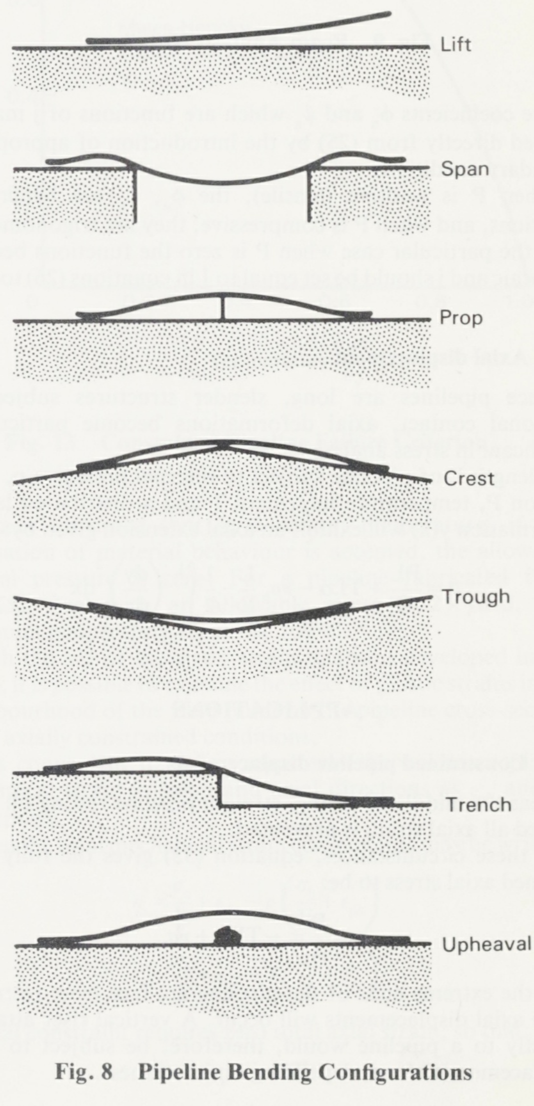


Fig. 8 Pipeline Bending Configurations

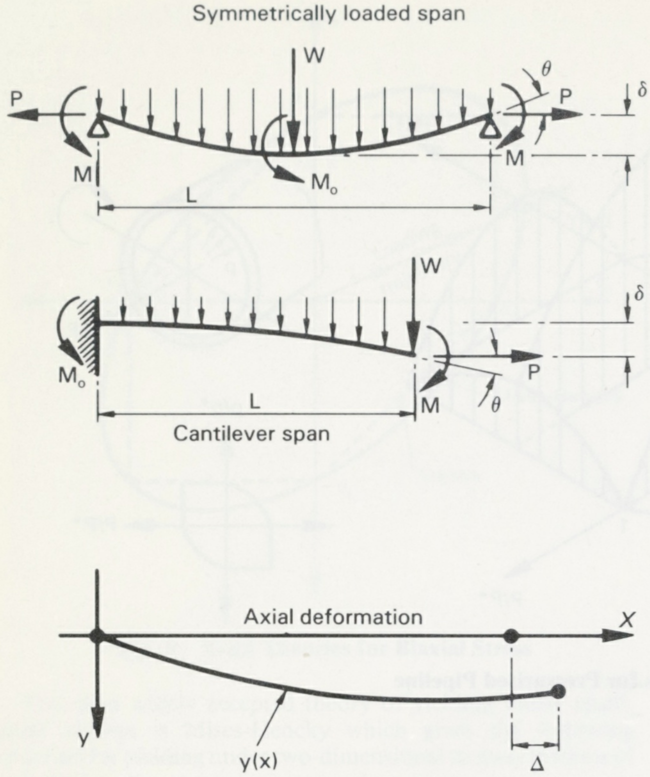


Fig. 9 Beam Analysis Notation

The coefficients ϕ_n and ψ_n which are functions of j may be derived directly from (25) by the introduction of appropriate boundary conditions.

When P is positive (tensile), the ϕ_n, ψ_n are hyperbolic functions, and when P is compressive, they are trigonometric.

In the particular case when P is zero the functions become algebraic and j should be set equal to 1 in equations (26) to (32).

4.2 Axial displacement

Since pipelines are long, slender structures subject to frictional contact, axial deformations become particularly significant in stress analysis.

A length L of pipeline subject to internal pressurre p, axial tension P, temperature increment T and a distributed lateral deformation y(x) will exhibit an axial extension given by:

$$\Delta = \frac{PL}{AE} + TL\alpha - \nu\sigma_h \frac{L}{E} - \frac{1}{2}\int_o^L \left(\frac{dy}{dx}\right)^2 dx \qquad (33)$$

5. APPLICATIONS

5.1 Constrained pipeline displacements

Near the middle of a long, straight pipeline resting on a flat seabed all axial strain is prevented.

In these circumstances, equation (33) gives the fully constrained axial stress to be:

$$\sigma_a = \frac{P}{A} = -TE\alpha + \nu\sigma_h \qquad (34)$$

If the extreme ends of the pipeline are free from restraint, large axial displacements will occur. A vertical riser attached directly to a pipeline would, therefore, be subject to large displacement or load, depending on its stiffness.

For this reason, a low stiffness expansion loop is introduced between pipeline and riser (*see* Fig. 1) in order to absorb the deformation without undue stress.

The temperature distribution along a submarine pipeline subject to convective heat loss may be described by:

$$T = T_1 e^{-kx} \qquad (35)$$

The constant k may be established either from a knowledge of material properties and convective conditions or, with more certainty, from information about temperatures at two distinct locations.

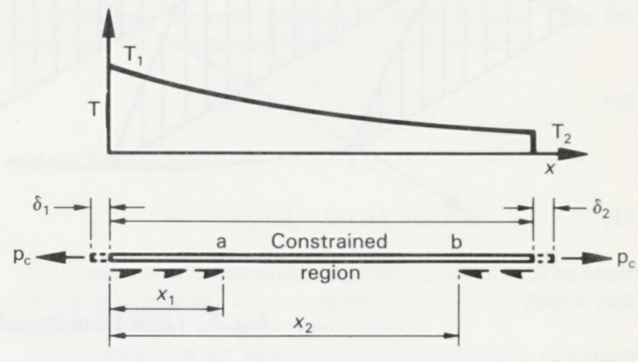


Fig. 10 Constrained Pipeline Notation

Frictional resistance, which always acts to oppose movement, is given by:

$$w_{fa} = \mu_a w_b \qquad (36)$$

where w_b = negative buoyancy
μ_a = axial friction coefficient

Therefore, axial force distribution along the first part of the pipeline from the input free end is given by:

$$P = w_{fa}x - Pc$$

where $P_c = \frac{\pi}{4} D_i^2 p$ = endcap pressure force

and axial strain is given by

$$\epsilon = -\frac{1}{AE}(w_{fa}x - P_c) + \alpha T_1 e^{-kx} - \nu\frac{\sigma_h}{E} \qquad (37)$$

The anchor point, x_1 from the free end, is the point along the pipeline at which displacement is zero. For a long line, this point will have zero strain and may be located by setting equation (37) to zero. However, x_1 cannot exceed half the length of the line, and this condition will override for short lines.

With reversed frictional resistance at the distant free end, axial force will be given by:

$$P = w_{fa}(L - x) - Pc$$

and the second anchor point is established by equating forces with those in the constrained region.

Displacements in both end regions may then be calculated by integration of axial strains. The procedures are illustrated by the results shown in Fig. 11 for a 12 inch line 5 km long subject to 170 bar pressure and a maximum temperature of 70°C.

The analysis, performed with the aid of the routine PIPEX, indicates the extent of the disturbed region (2.38 km), with maximum displacement of 1.21 m.

The non-linear nature of the problem is illustrated in Fig. 12 for a similar pipeline uniformly heated to a range of temperatures.

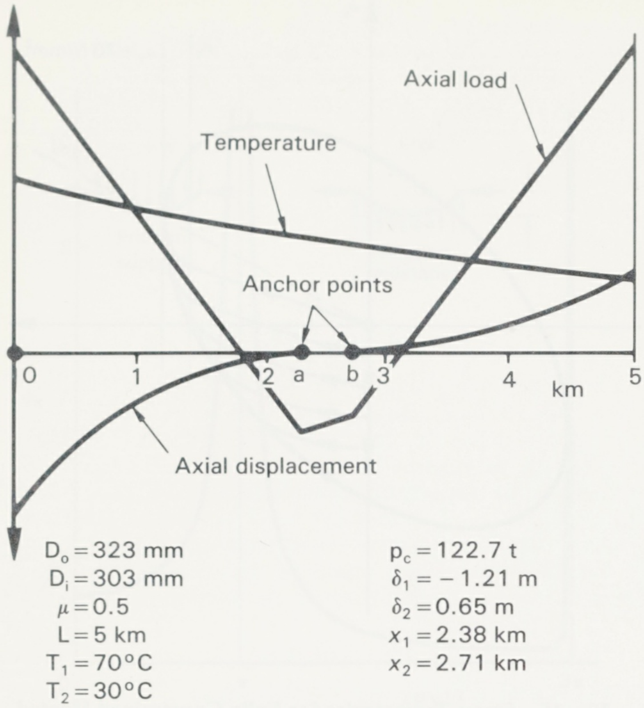


Fig. 11 Force and Displacement Distributions for a Heated Pressureised Pipeline

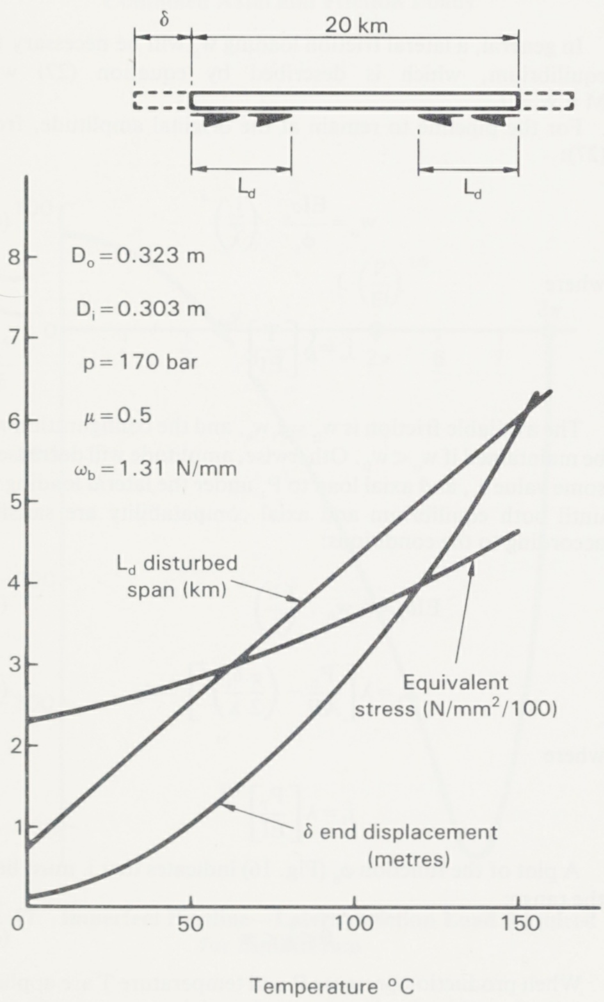


Fig. 12 Temperature Effects on a Pressurised Pipeline

5.2 Constrained pipeline strength

A fully constrained pipeline under hoop stress σ_h and temperature increment T will experience an axial stress σ_a given by equation (34).

By substitution into the yield criterion given by equation (12), it is possible to calculate the maximum hoop stress that can be allowed for any given temperature if unlimited yielding is to be avoided. This is found by solving the following quadratic equation:

$$(1-\nu+\nu^2)\left(\frac{\sigma_h}{\sigma_y}\right)^2+(1-2\nu)\left(\frac{\sigma_h}{\sigma_y}\right)\left(\frac{E\alpha T}{\sigma_y}\right)+\left(\frac{E\alpha T}{\sigma_y}\right)^2-1=0 \tag{38}$$

This is plotted in Fig. 13, together with the hoop stress limitation.

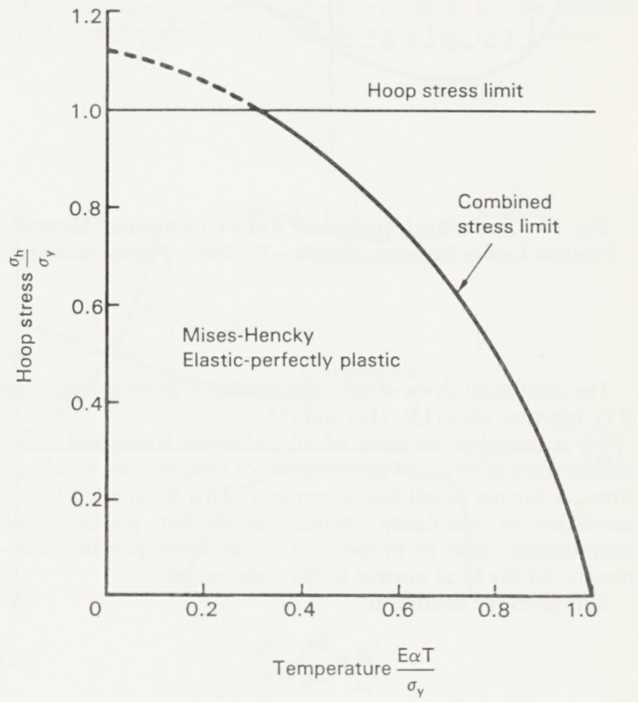


Fig. 13 Consrained Pipeline Failure Criterion

It may be noted that at a certain temperature, if this idealisation of material behaviour is accepted, the allowable internal pressure is zero. For a pipeline fabricated from API X52 steel with an allowable stress 0.72×yield, this maximum temperature would be 109°C.

With the aid of the stress-strain relations developed in 3.1 above, it is possible to examine the effect of plastic strains in the neighbourhood of the yield point on the pipeline cross-section under axially constrained conditions.

It is convenient to define separately the unknown plastic mechanical strain in hoop and axial directions as ϵ_{ph} and ϵ_{pa} respectively.

Then the total mechanical strain components may be written:

$$\epsilon_a=\frac{\sigma_a}{E}+\epsilon_{pa}-\nu\left(\frac{\sigma_h}{E}+\epsilon_{ph}\right) \tag{39}$$

$$\epsilon_h=\frac{\sigma_h}{E}+\epsilon_{ph}-\nu\left(\frac{\sigma_a}{E}+\epsilon_{pa}\right) \tag{40}$$

Axial constraint requires zero total axial strain, so that:

$$\epsilon_a=-\alpha T \tag{41}$$

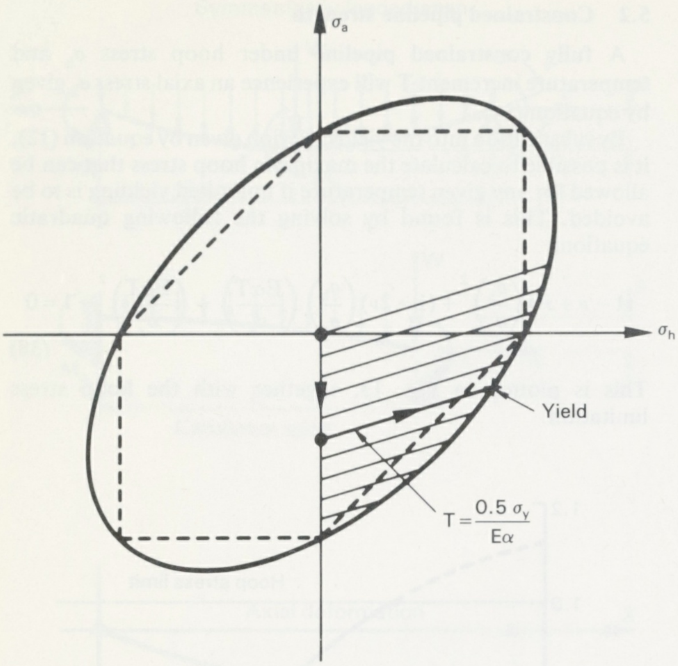


Fig. 14 Stress Trajectories for Fully Constrained Heated Pipeline Under Pressure Elastic—Perfectly Plastic material

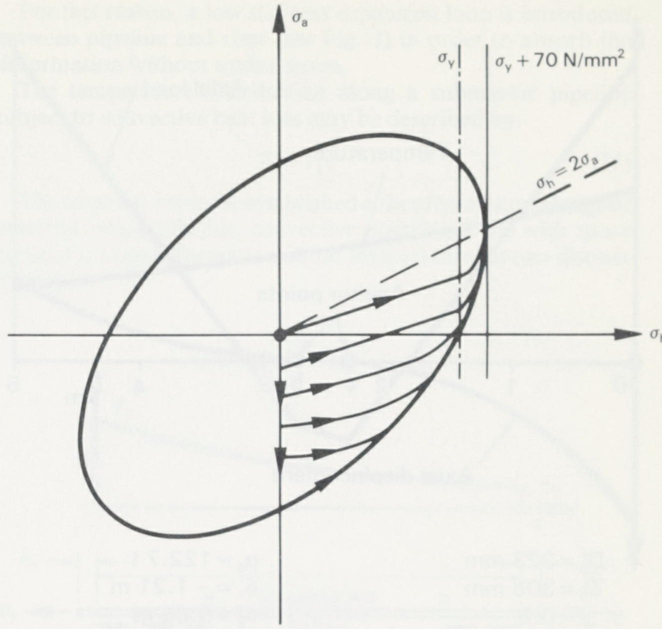


Fig. 15 Stress Trajectories for Fully Constrained Heated Pipeline Under Pressure Work Hardening Material

The multiaxial stress-strain relationship is given by equation (11), together with (13), (14) and (15).

For a complete solution of all unknown stress and strain components in terms of given values of temperature and hoop stress, a further condition is required. This is supplied by the condition of normality, which states that plastic strain components occur in proportion to the corresponding components of the local normal to the yield surface.

This gives the condition:

$$\frac{\epsilon_{pa}}{\epsilon_{ph}} = \frac{2\sigma_a - \sigma_h}{2\sigma_h - \sigma_a} \tag{42}$$

The above relationships may be incorporated in a numerical procedure which allows the loading trajectory, in terms of stresses, to be traced for constrained pipeline sections.

Fig. 14 indicates the path followed by the elastic-perfectly plastic solution, each trajectory terminating at the yield surface.

Fig. 15 shows how strain hardening allows the equilibrium of significantly greater hoop stresses, with trajectories converging towards a 2 : 1 hoop to axial stress ratio.

5.3 Pipeline imperfections

The above analysis of constraint effects was based on the assumption that the pipeline is laid in a precise straight line. In real life, nothing is perfectly straight, and pipelines are no exception. In order to obtain some indication of the effects of laying imperfections, a pipeline laid along a sinusoidal path has been considered.

The initial path can be adequately described by just two quantities, wavelength and amplitude. A residual laying tension may be present, but otherwise the pipeline may be considered to be unstressed. The initial shortening, the amount by which the length of one half wavelength of pipe exceeds the distance between adjacent nodes, is given by:

$$\Delta_o = \lambda\left[\frac{Te}{AE} - \left(\frac{\pi}{2}\frac{\delta_o}{\lambda}\right)^2\right] \tag{43}$$

In general, a lateral friction loading w_o will be necessary for equilibrium, which is described by equation (27) with $M = W = 0$.

For the pipeline to remain at the original amplitude, from (27):

$$w_o = \frac{EI\sigma_o}{\phi_6} \cdot \left(\frac{j_o}{\lambda}\right)^4 \tag{44}$$

where

$$j_o = \lambda\left[\frac{T}{EI}\right]^{1/2}$$

The available friction is $w_{ft} = \mu_t w_b$, and the configuration will be maintained if $w_o < w_{ft}$. Otherwise, amplitude will decrease to some value δ_1, and axial load to P_1 under the lateral loading w_{ft} until both equilibrium and axial compatability are satisfied according to the conditions:

$$EI\delta_1 = \phi_6 w_{ft} \cdot \left(\frac{\lambda}{j_1}\right)^4 \tag{45}$$

$$\Delta_1 = \lambda\left[\frac{P_1}{AE} - \left(\frac{\pi}{2}\frac{\delta_1}{\lambda}\right)^2\right] = \Delta_o \tag{46}$$

where

$$j_1 = \lambda\left[\frac{P_1}{EI}\right]^{1/2}$$

A plot of the function ϕ_6 (Fig. 16) indicates that j_1 must lie in the range:

$$0 < j_1 < \pi \tag{47}$$

When production pressure P and temperature T are applied, amplitude will tend to increase as the axial compressive force in the pipeline increases.

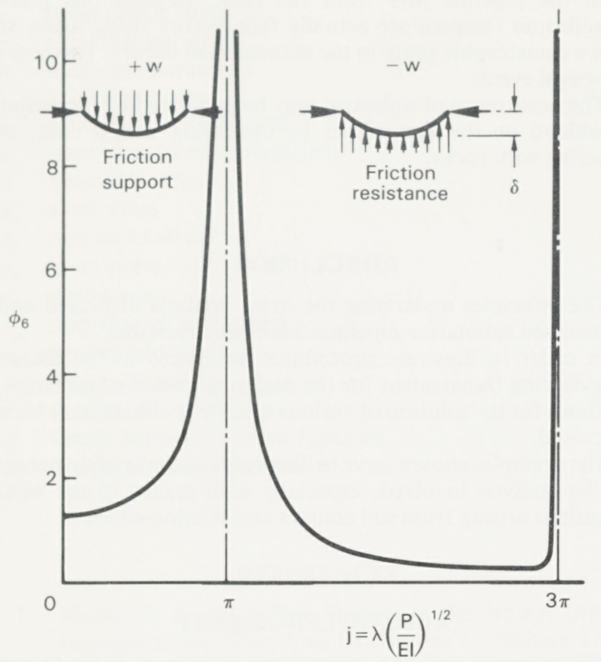


Fig. 16 imperfect Pipeline—Lateral Displacement Due to Combined Axial and Friction Loads

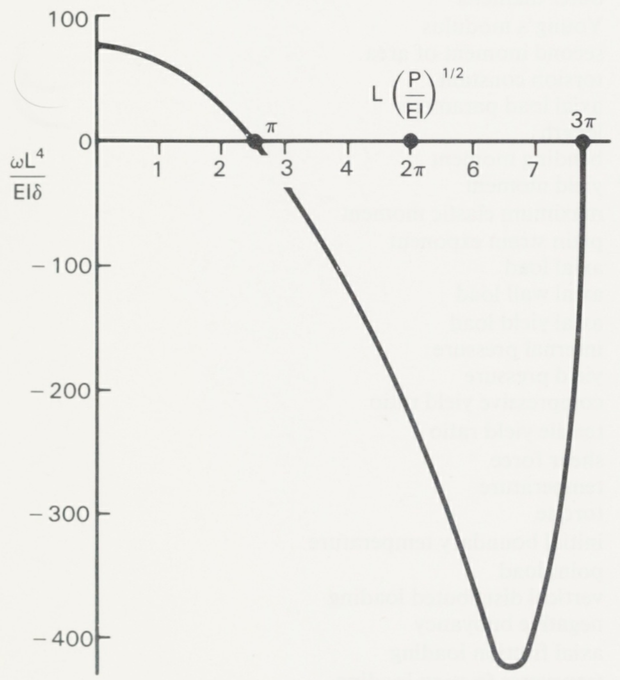


Fig. 17 Imperfect Pipeline—Lateral Friction Load Required for Equilibrium

Frictional resistance will be mobilised and, for equilibrium to be achieved, the axial load parameter j must be contained within the range:

$$\pi < j < 3\pi$$

which may be noted from Fig. 16.

The lateral resistance required for equilibrium at any given amplitude is found from (45) to be:

$$W = \frac{EI}{\phi_6} \cdot \delta \cdot \left[\frac{j}{\lambda}\right]^4 \tag{48}$$

This is plotted in Fig. 17. Clearly the lateral resistance may not be sufficient for equilibrium at the original imperfection wavelength, therefore, the original wavelength must change in order to reach a stable state.

The process of finding a solution on this basis is illustrated in Fig. 18, which indicates how the simultaneous satisfaction of compatibility and equilibrium at the largest wavelength is achieved. This procedure is incorporated in the routine PIPAN3 from which the results shown in Fig. 19 are taken.

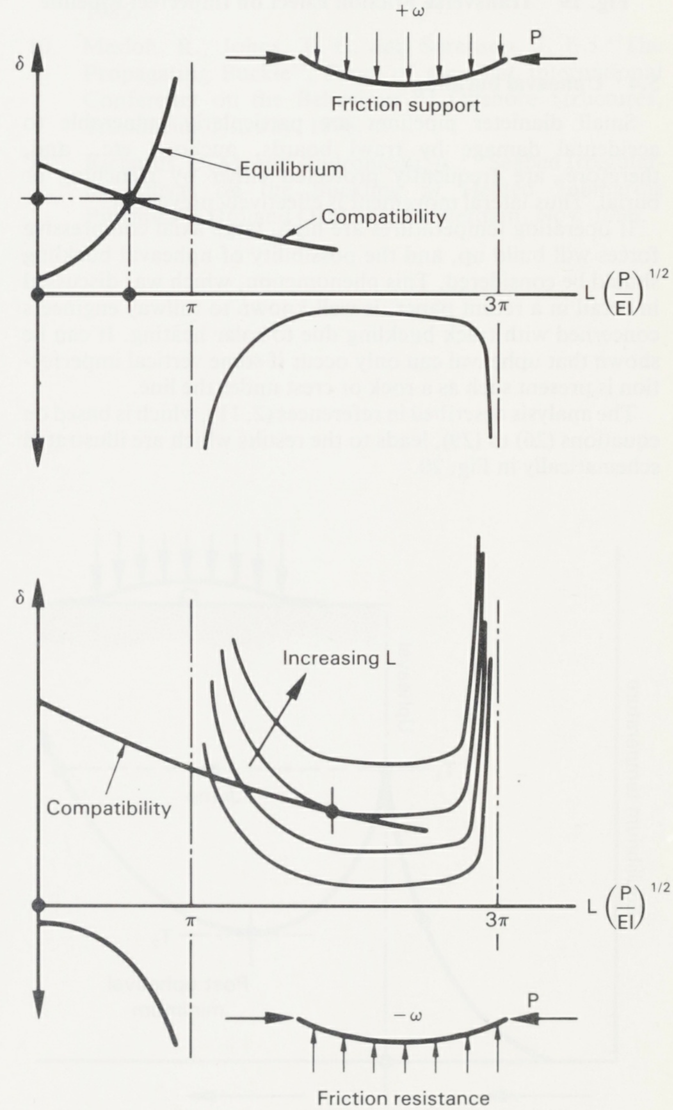


Fig. 18 Imperfect Pipeline Solution Criteria

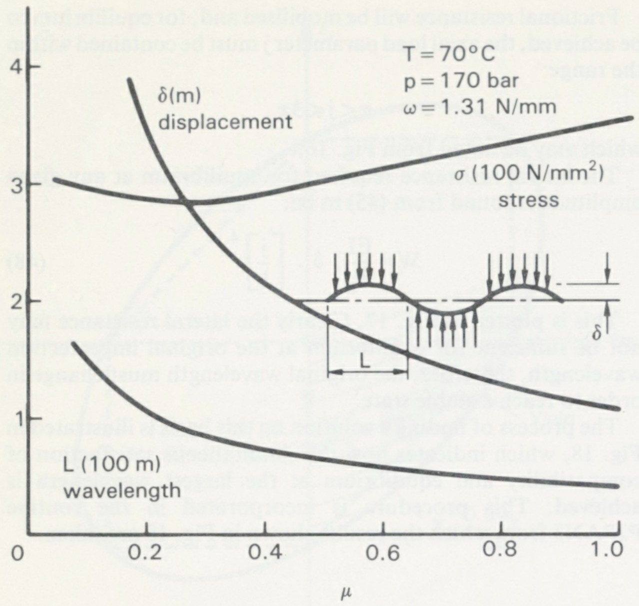


Fig. 19 Transverse Friction Effect on Imperfect Pipeline

5.4 Upheaval buckling

Small diameter pipelines are particularly vulnerable to accidental damage by trawl boards, anchors, etc., and, therefore, are frequently protected either by trenching or burial. Thus lateral movement is effectively prevented.

If operating temperatures are high, large axial compressive forces will build up, and the possibility of upheaval buckling should be considered. This phenomenon, which was discussed in detail in a recent paper, is well-known to railway engineers concerned with track buckling due to solar heating. It can be shown that upheaval can only occur if some vertical imperfection is present such as a rock or crest under the line.

The analysis described in references (2, 11), which is based on equations (26) to (29), leads to the results which are illustrated schematically in Fig. 20.

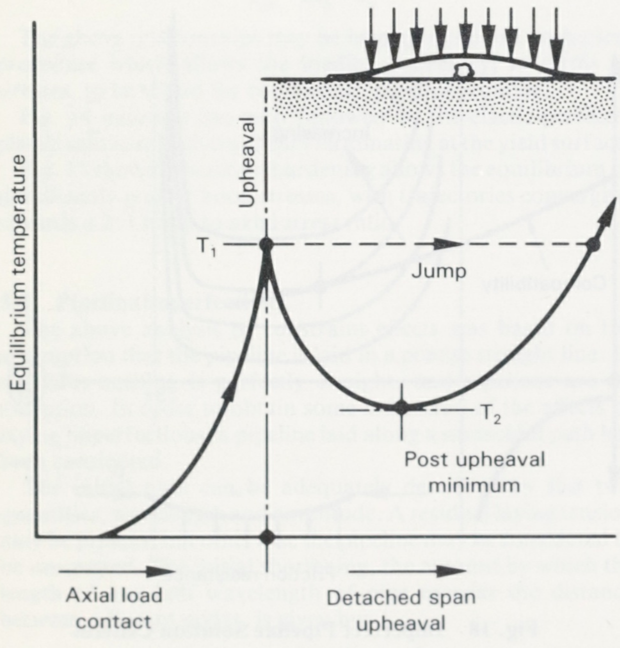


Fig. 20 Upheaval Buckling

At each stage of loading the pipeline temperature necessary for equilibrium can be calculated. In the case of the prop imperfection under the line, equilibrium temperature increases until the pipeline lifts from the rock. Beyond this point equilibrium temperature actually falls before rising again so that a catastrophic jump in the deformation occurs. This is the upheaval event.

The occurrence of upheaval may be prevented by increasing download on the line, either by increasing weight coat, or covering with rocks.

6. CONCLUSION

The principles underlying the stress analysis of heated and pressurised submarine pipelines have been reviewed.

In order to illustrate procedures developed in the Ocean Engineering Department for the design appraisal of pipelines, methods for the solution of various specific problems have been discussed.

The examples chosen serve to illustrate the challenging nature of the analysis involved, especially with regard to the non-linearities arising from soil contact and friction effects.

ACKNOWLEDGEMENT

The author would like to thank his colleagues in the Ocean Engineering Department for their help in preparing this paper.

NOTATION

A	area
a	plastic strain coefficient
D_i	inner diameter
D_o	outer diameter
E	Young's modulus
I	second moment of area
J	torsion constant
j	axial load parameter
L	length
M	bending moment
M*	yield moment
M_e^*	maximum elastic moment
n	plain strain exponent
P	axial load
P_w	axial wall load
P*	axial yield load
p	internal pressure
p*	yield pressure
r_c	compressive yield ratio
r_t	tensile yield ratio
S	shear force
T	temperature
T_o	torque
T_1	initial boundary temperature
W	point load
w	vertical distributed loading
w_b	negative buoyancy
w_{fa}	axial friction loading
w_{ft}	transverse friction loading
x	axial coordinate
y	transverse coordinate
α	temperature coefficient
Δ	axial displacement
δ	transverse displacement
ϵ_c	curvature strain

ϵ_e equivalent strain
ϵ_p axial pressure strain
ϵ_{pa} axial plastic strain
ϵ_{pn} hoop plastic strain
ϵ_t axial thermal strain
θ load ratio parameter
λ axial shortening
μ_a axial friction coefficient
μ_t transverse friction coefficient
ν Poisson's ratio
σ_a axial stress
σ_e equivalent stress
σ_h hoop stress
σ_y yield stress
σ_{yc} compressive yield stress
σ_{yt} tensile yield stress
$\sigma_{1,2}$ principal stresses
τ shear stress
ϕ_n simply supported beam function
ψ_n cantilever beam function

REFERENCES

1. Mousselli, A. H.: "Pipe Stresses at the Seabed During Installation and Trenching Operations", Offshore Technology Conference, OTC 2965, Houston, May 1977.
2. Boer, S., Hulsbergen, C. H., Richards, D. M., Klok, A. and Biaggi, J. P.: "Buckling of High Temperature Oil Line Covered by Gravel", Offshore Technology Conference, OTC 5294, Houston, May 1986.
3. Kerr, A. D.: "On the Stability of the Railroad Track in the Vertical Plane", Rail International, Vol. 5, Feb. 1974, pp 131–142.
4. Timoshenko, S. P. and Gere, J. M.: "Theory of Elastic Stability", 2nd ed., McGraw-Hill, New York, 1961.
5. Richards, D. M.: "Seabed Topography Effects on Pipeline Configuration and Strength", presented at Offshore Goteborg '83, Gothenburg, Sweden, March 1983.
6. Courbon, J.: "The Equilibrium of a Heated Beam Resting over a Rigid Horizontal Plane", (in French), Journal de Mecanique Appliquee, Vol. 4, No. 4, 1980.
7. Yassine, E. M. and Tchernikine, V. I.: "Stabilite des Canalisations Enterees", Editions Niedra-Moscou, 1968.
8. Jensen, J. J.: "ISH Manual for the Static and Dynamic Analysis of Three-Dimensional Frames", Dept. of Ocean Engineering, The Technical University of Denmark, Lyngby, Denmark, October 1979.
9. Winter, P. E., Stark, J. W. B. and Witteveen, J.: "Collapse Behaviour of Submarine Pipelines", Chapter 7 of Shell Structures Stability and Strength, edited by Narayanan, K., Elsevier Applied Science Publishers, 1985.
10. Mesloh, R., Johns, T. G. and Sorenson, J. E.: "The Propagating Buckle", Proc. of the First International Conference on the Behaviour of Offshore Structures, Trondheim, Norway, 1976.
11. Richards, D. M. and Andronicou, A.: "Seabed Irregularity Effects on the Buckling of Heated Submarine Pipelines", Holland Offshore, Amsterdam, Nov. 1986.

ε_e equivalent strain
ε_p axial pressure strain
ε_{ap} axial plastic strain
ε_{hp} hoop plastic strain
ε_{T} axial thermal strain
θ load ratio parameter
λ axial shortening
μ_a axial friction coefficient
μ_t transverse friction coefficient
ν Poisson's ratio
σ_a axial stress
σ_e equivalent stress
σ_h hoop stress
σ_y yield stress
σ_{yc} compressive yield stress
σ_{yt} tensile yield stress
σ_{h}^{*} equivalent hoop stress
τ hoop shear stress
ϕ_s simply supported beam function
ϕ_c cantilever beam function

REFERENCES

1. Mousselli, A. H.: "Pipe Stresses at the Seabed During Installation and Trenching Operations", Offshore Technology Conference, OTC 2965, Houston, May 1977.
2. Ellinas, C. P., Hobbs, R. E., Richards, D. M., Klever, A. and Boggs, J. P.: "Buckling of High Temperature Oil Line Covered by Gravel", Offshore Technology Conference, OTC 5294, Houston, May 1986.
3. Kerr, A. D.: "On the Stability of the Railroad Track in the Vertical Plane", Rail International, Vol. 5, Feb. 1974, pp. 131–142.
4. Timoshenko, S. P. and Gere, J. M.: "Theory of Elastic Stability", 2nd ed., McGraw-Hill, New York, 1961.
5. Kolstad, O. M.: "Seabed Topography Effects on Pipeline Configuration and Stresses", presented at Offshore Goteborg '83, Gothenburg, Sweden, March 1983.
6. Combes, J.: "The Equilibrium of a Heated Beam Resting over a Rigid Horizontal Plane", (in French), Journal de Mécanique Appliquée, Vol. 4, No. 4, 1980.
7. Yaremov, P. M. and Tchernikine, V. I.: "Stabilité des Canalisations Enterrées", Editions Nedra, Moscow, 1985.
8. Jensen, J. J.: "ISH Manual for the Static and Dynamic Analysis of Three Dimensional Frames", Dept. of Ocean Engineering, The Technical University of Denmark, Lyngby, Denmark, October 1977.
9. Winter, P. E., Stark, J. W. B. and Witteveen, J.: "Collapse Behaviour of Submarine Pipelines", Chapter 7 of Shell Structures Stability and Strength, edited by Narayanan, R., Elsevier Applied Science Publishers, 1985.
10. Mesloh, R., Johns, T. G. and Sorensen, J. E.: "The Propagating Buckle", Proc. of the First International Conference on the Behaviour of Offshore Structures, Trondheim, Norway, 1976.
11. Richards, D. M. and Andronicou, A.: "Seabed Irregularity Effects on the Buckling of Heavily Restrained Pipelines", Holland Offshore, Amsterdam, Nov. 1986.

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STRENGTH ANALYSIS OF SELF ELEVATING UNITS

W. J. Winkworth

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Hon. Sec. C. M. Magill
71 Fenchurch Street, London, EC3M 4BS

STRENGTH ANALYSIS OF SELF ELEVATING UNITS

by

W. J. Winkworth

Prior to joining Lloyd's Register Walter Winkworth first gained experience in the petro-chemical industry working on the structural design and analysis of plant pipework for Kellogg International Corporation. Following this he moved to the British Aircraft Corporation, Weybridge, where he was engaged as a stress engineer on projects such as Concorde and the Tornado.

In 1971 he joined the Ocean Engineering Department of Lloyd's Register as a Structural Engineer. Since then he has performed design appraisal analyses on many different types of offshore structures including fixed steel platforms, semi-submersibles and jack-ups. He has also been involved in the development of the computer programs and analysis techniques used by the Society in this work. In 1978 he was appointed Deputy Head of the Ocean Engineering Department.

SYNOPSIS

This paper describes the Society's approach to design appraisal and approval of jack-ups, particularly the leg structure. As part of the approval of any offshore unit the Society performs an independent structural analysis. The assumptions on which the structural analysis of a jack-up is based and the methods used are described. In addition some of the design considerations applicable to the new generation of deep water, harsh environment jack-ups are discussed.

TABLE OF CONTENTS

1. INTRODUCTION

Jack-up platforms are the most common type of offshore mobile unit and are the workhorse of the offshore exploration industry. In recent years jack-ups have also been used as crane barges, accommodation units and as early production platforms.

There are several different designs of jack-up built to date. For many shallow water units the legs are formed by a single member, either square or round, the method of jacking being a hydraulic positive engagement system which utilizes pins that are moved step by step up and down the leg. There are also mat type units where the legs are connected at the bottom by a large structural mat.

The jack-up considered in this paper, however, is the most common type of unit used for deep water. It will have either three or four legs, each leg constructed of a triangular or square lattice framework. At the bottom of the legs are large load

bearing shells called 'spud cans' which rest on the seabed when the unit is jacked up. The legs are moved up and down through the hull utilizing a rack and pinion jacking system. Each corner or chord of the lattice leg has one or two racks attached to it. The pinions are housed in the jacking towers on the deck. A typical system will consist of six pinions at each corner of the leg. For a triangular leg this is a total of 18 pinions, each driven by an electric or hydraulic motor. Each pinion is able to jack a maximum load of 200 to 300 tonnes. A typical jacking speed for these type of structures is of the order of half a metre per minute.

2. OPERATION AND CRITICAL DESIGN CONDITIONS

2.1 Pre-loading

While under tow the legs of the jack-up are raised as far as possible to minimise drag. At location, the legs are lowered through the hull until the spud cans reach the seabed and the hull is then jacked clear of the water and the legs pre-loaded. The Society's Rules[1] require that every jack-up should have 100% pre-load capability. That is, a load equivalent to the maximum gravity plus the maximum storm loading must be applied to each leg. This is achieved either by variable ballast or by differential jacking on the legs. The purpose of this exercise is to test the foundation and to ensure a firm footing. The hull is then jacked up clear of the water such that there is an air gap between the hull and the largest expected wave occuring at high tide.

2.2 Critical Design Conditions

Three important design conditions must always be considered:

(i) On location: under a realistic combination of extreme environmental conditions in the fully jacked-up configuration. Since the member sizes in the leg may be reduced for operation in shallower water, several water depths must be investigated.

(ii) In transit: with the maximum length of legs elevated above the barge for tow under the maximum specified motions.

(iii) Jacking-up: during maximum allowable waves, the impact of the legs on the seabed may be a design case.

The above conditions are critical primarily for the legs and their back-up structure. The general barge structure, decks etc., can be most highly stressed during the floating phase and by local loading cases and are checked according to the Society's *Rules for the Construction and Classification of Mobile Offshore Units*. The hull is constructed from normal ship grade steel except at critical locations around the jacking towers and leg guides where increased notch toughness and ductility are required. It is essential to keep the leg weight to a minimum in order to improve the floating stability during tow with the legs fully jacked up and to reduce the loads in the leg, guides and jacking tower. Therefore the steels used for the leg construction are generally of very high strength having yields of up to 690.0 N/mm^2.

Only the analysis of the jacked-up condition on location is considered in this paper. However, the general analysis method is applicable to the other load cases. The analysis can be divided into the following aspects:

(a) Environment
(b) Determination of loading
(c) Overall analysis
(d) Detailed analysis

3. ENVIRONMENT

3.1 General

Jack-up platforms are essentially mobile platforms and therefore the environmental conditions considered in the design are normally specified by the owner. The operation of the unit is then restricted to those areas where the extreme storm conditions are equal to or less than the design conditions considered. Where the area of operation is known the Society would expect to approve the environmental conditions chosen for design.

3.2 Importance of Current

It is not generally appreciated that current makes a very large contribution to the total loading on lattice-leg type jack-ups. Typical results, illustrating the increase in overall wave loading with variations in current, are given in Table 1.

Table 1 Wave loading due to current on lattice legs

Water Depth (m)	Wave Height (m)	Horiz. Force Zero Current (tonnes)	Horiz. Force 0.6 m/s Current (tonnes)	Horiz. Force 1.2 m/s Current (tonnes)	Horiz. Force 1.8 m/s Current (tonnes)
78	23	1032	1189	1407	1646
91	20	669	810	1006	1225
110	17	430	539	717	920

3.3 Defining Environmental Parameters

The number of design cases defined in the Operations Manual does not normally cover all the combinations of wind, wave, current and water depth to which a jack-up may be exposed. There is therefore a need for further guidance in order to determine safe operating locations for the unit. One of the most convenient ways to display this information is in the form of a carpet plot, an example of which is shown in Fig. 1. Graphs such as the one illustrated should be included in the Operations Manual.

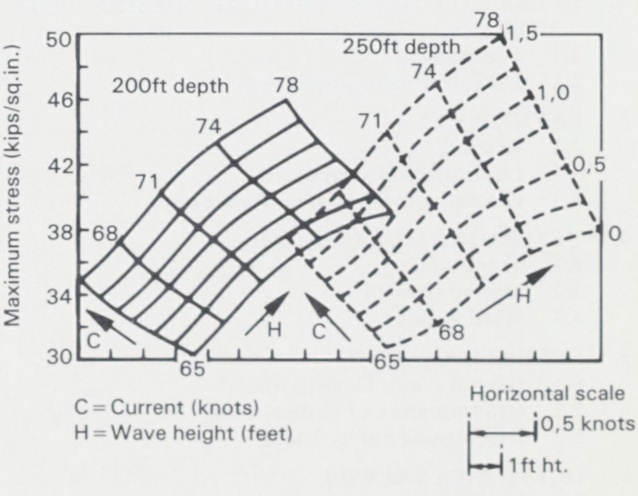

Fig. 1 Maximum stress in a typical jack-up leg due to waves and currents

4. DETERMINATION OF LOADS

4.1 Wave Loading

Critical to both the wind load and wave and current loads is the evaluation of the drag coefficient for the leg chord. The shape of the leg chord carrying the rack is generally not circular. There is very little published data on drag coefficient for shapes typical of jack-up leg chords, although some has recently been published for a specific chord design.(2) Because of this lack of data the Society has conducted a series of wind tunnel experiments to determine comparative results for most of the major types of leg chords presently in use. These tests were performed in an 8 ft diameter wind tunnel with an effective chord diameter of 8 ins.

Within the scope of this paper there is insufficient space to deal fully with the results of these tests, however a typical set of results is given in Fig. 2. These indicate how the drag coefficient varies with orientation of the chord to the flow and with Reynolds number. For the braces, which are generally circular, a C_D of 0.6 is assumed.

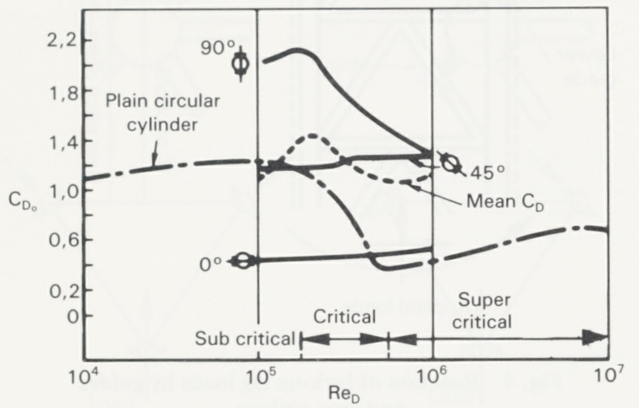

Fig. 2 Variation of drag coefficient with Reynolds number

4.2 Marine Growth

Traditionally, the effect of marine growth has not been included in the design of jack-ups. The basis of this assumption is that marine growth is discouraged by the frequent location moves and because the legs are in air for considerable periods during tow to new locations. In addition, there is ample opportunity for cleaning during the tow. However, where a unit will be operating for a long period at one location, particularly alongside a fixed platform which provides a source of 'infection', an allowance for marine growth should be included in the wave load calculations.

4.3 Wind Loads

It should be noted that the wind loading can alter significantly with water depth. This is because, at reduced water depths, the length of exposed leg is increased and vice versa. Formulae for calculating wind loading are given in the Society's *Rules for Mobile Offshore Units*.

Wave loads are calculated using the Society's in-house wave programs and, combined with wind and gravity loads, are applied to the overall jack-up model.

5. OVERALL ANALYSIS

5.1 Overall Finite Element Model

In this idealisation, it is not usually necessary to spend a great deal of effort in accurately representing the stiffness of the barge as this is very stiff in comparison with the legs. The attachment of the legs to the hull is more critical and the idealisation of this area is carefully assessed. Based on the expected penetration at site, the legs are normally assumed pinned at 3 metres below the mudline. A typical finite element model is illustrated in Fig. 3.

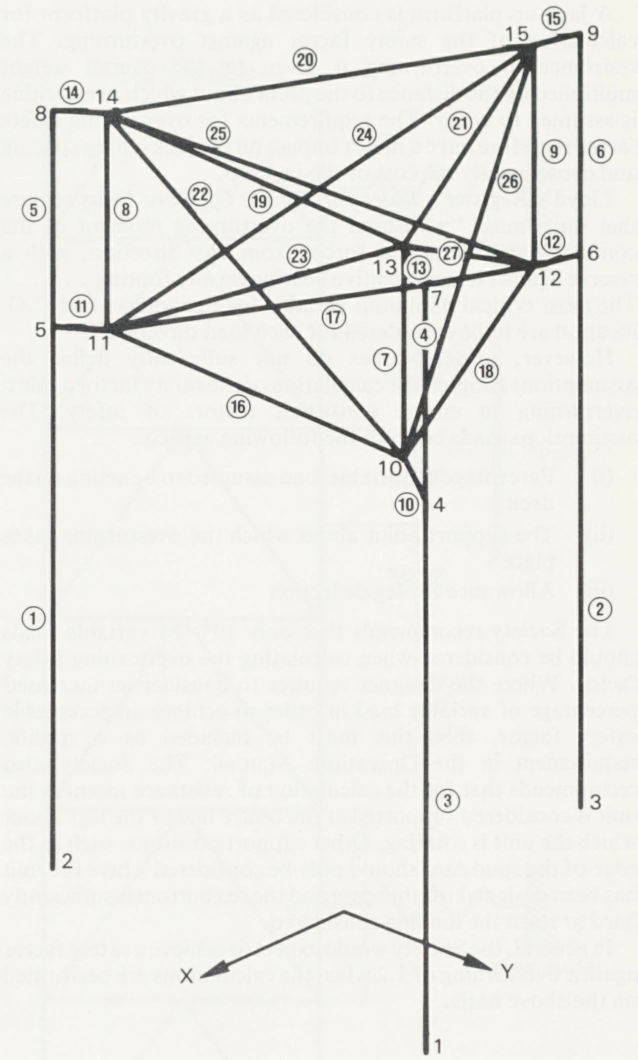

Fig. 3 Typical overall jack-up NASTRAN model

The secondary moment due to leg deflection is included in the internal loads at the lower guide.

The overall stability of the leg acting as a beam column is then checked using the results of this analysis.

5.2 Determination of Critical Case

The selection of the critical case for a jack-up leg design is dependent on determining the combination of bending moment and axial load, together with the appropriate local loading from the guides and pinions, that produces the maximum stresses in the leg structure. Different cases may be critical for the chord, horizontal brace and diagonal brace. A rigorous search for the critical conditions would require a finite element analysis for each wave direction and wave crest position. This would have to be carried out at each water depth and for at least two positions of guide location relative to leg bracing. Since this approach is both prohibitively expensive and time consuming, engineering

judgement and experience are used to determine the critical cases. In general, the Society has found that for a 3-leg unit, the wave approach direction and crest position giving the maximum axial load for the leg structure is critical.

5.3 Overturning Safety Factor

A jack-up platform is considered as a gravity platform for calculation of the safety factor against overturning. The resistance to overturning is given by the overall weight multiplied by the distance to the point about which overturning is assumed to occur. The requirements for overturning safety factor therefore have a major impact on the jack-up leg spacing and consequently the cost of the jack-up.

Lloyd's Register's *Rules for Mobile Offshore Units* require that units must "withstand the overturning moment of the combined environmental forces from any direction, with a reserve against loss of positive bearing on any footing . . . . . . The most critical minimum variable load condition and C.G. location are to be considered for each load direction".

However, present codes do not sufficiently define the assumptions adopted for calculation of the safety factor against overturning to ensure consistent factors of safety. The assumptions made concern the following aspects:

(i) Percentage of variable load assumed to be acting on the deck

(ii) The support point about which the overturning takes place

(iii) Allowance for leg deflection

The Society recommends that only 10% of variable loads should be considered when calculating the overturning safety factor. Where the designer requires to consider an increased percentage of variable load in order to achieve an acceptable safety factor, then this must be included as a specific requirement in the Operations Manual. The Society also recommends that for the calculation of resistance moment the unit is considered supported at the centre line of the legs about which the unit is rotating. Other support positions, such as the edge of the spud can, should only be considered where the unit has been designed for that case and the sea bottom is sufficiently hard to resist the loading considered.

In general, the Society would expect to achieve a safety factor against overturning of 1.2 when the calculations are performed on the above basis.

6. DETAILED ANALYSIS

6.1 Idealisation

It is necessary to check in detail the maximum stresses in the leg chord and leg bracing. These generally occur at the leg to hull connection. The jack-house, leg guides and back-up structure are also checked at this point in the analysis.

The jack-up leg considered is attached to the hull by guides and gear pinions. The loads in the leg are reacted in the following manner; shear is reacted by the guides and vertical load by the pinions or jacks. For a jack-up with a rack and pinion jacking system the bending moment may be reacted by differential horizontal shears or by varying vertical reactions at the pinions. The experience of the Society is that the pinion load path is relatively stiff and will generally attract load up to the limit of the pinion brake capacity. The pinions do not normally provide for a reverse loading reaction, that is, where the differential vertical load at one chord due to the overall leg bending moment is larger than the gravity load. The overall reaction system is shown in Fig. 4.

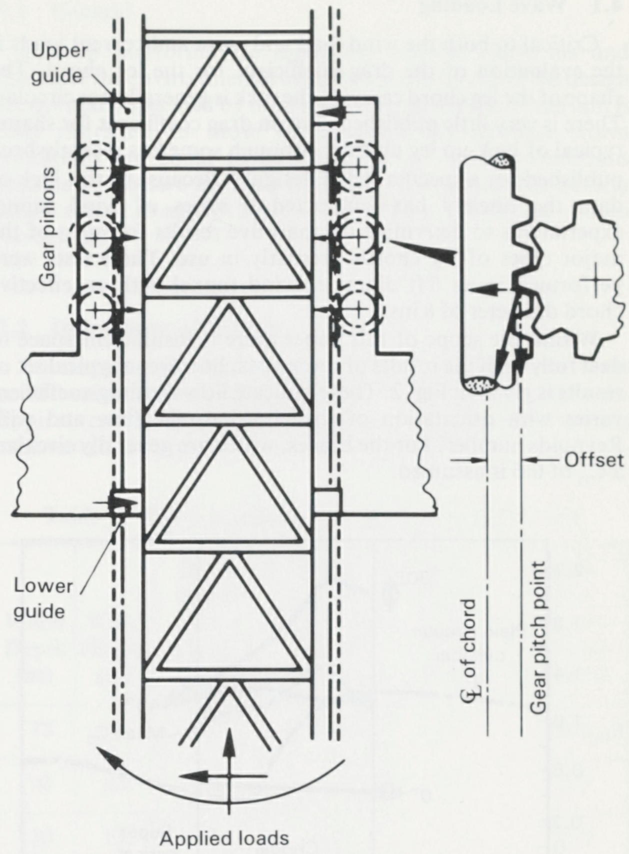

Fig. 4 Reaction of jack-up leg loads by guides and gear pinions

The applied shear, axial load and bending moment in the leg are determined from the overall analysis. In the detailed finite element analysis these loads are applied to the leg several bay lengths below the lower guide to ensure the correct distribution of internal loads.

For designs on which some form of leg clamping device is fitted, reverse vertical loading is allowed.

The stiffness of the guides, pinions and back-up structure at both the pinions and the upper and lower guides is represented by springs. The springs must take account of the stiffness of the jacking system and whether it is a fixed system or a floating system supported on shock absorbers.

Where the chord has a rack on one side only, the rack and pinion type jacking system produces two other important local loading effects:

(i) The offset of the gear pitch point from the chord centre line causes a local bending moment in the chord.

(ii) The angle of the rack teeth produces a side component to the vertical load.

A further complication is that the guides support the leg chords in a normal direction only and this has to be simulated in the analysis. The effect of the guide system on the way in which the load in the leg is reacted is shown in Fig. 5(a). Method 1 shows a circular guide acting against the leg chord. For some directions of loading the reaction is on one chord only and this is obviously important in the structural design. Method 2 involves guides acting on the rack or on side plates. It can be seen that the pattern of load in the leg will vary considerably depending on the guide system adopted.

A typical idealisation of this area is illustrated in Fig. 5(b) (shown two-dimensional for clarity). The following should be noted:

(a) Nodes 1, 2 and 3 are released for all but vertical loads.

(b) Nodes 4 and 5 represent the guides and must be released appropriately for both load and direction depending on the leg chord and loading direction considered.

(c) In most cases, there is a cut-off limit to the vertical load transmitted via the gear pinions. When the critical pinion reaches this limit, further increases in overall loads are distributed to the other pinions. Several iterations may be required to determine the final internal load distribution.

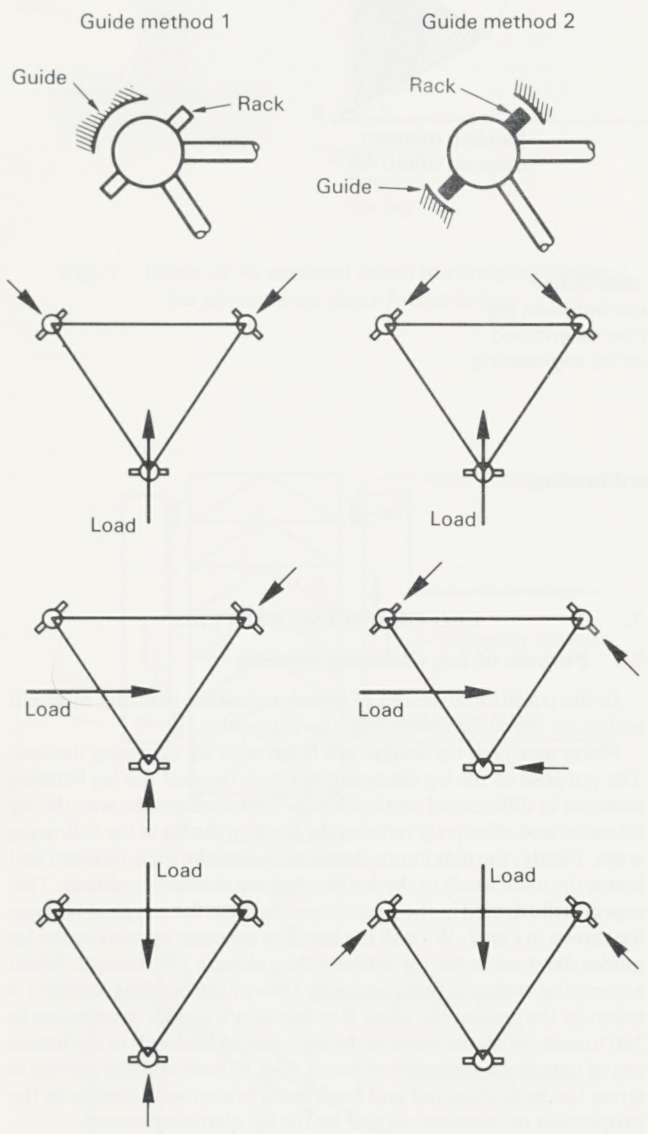

Fig. 5(a) Guide reactions

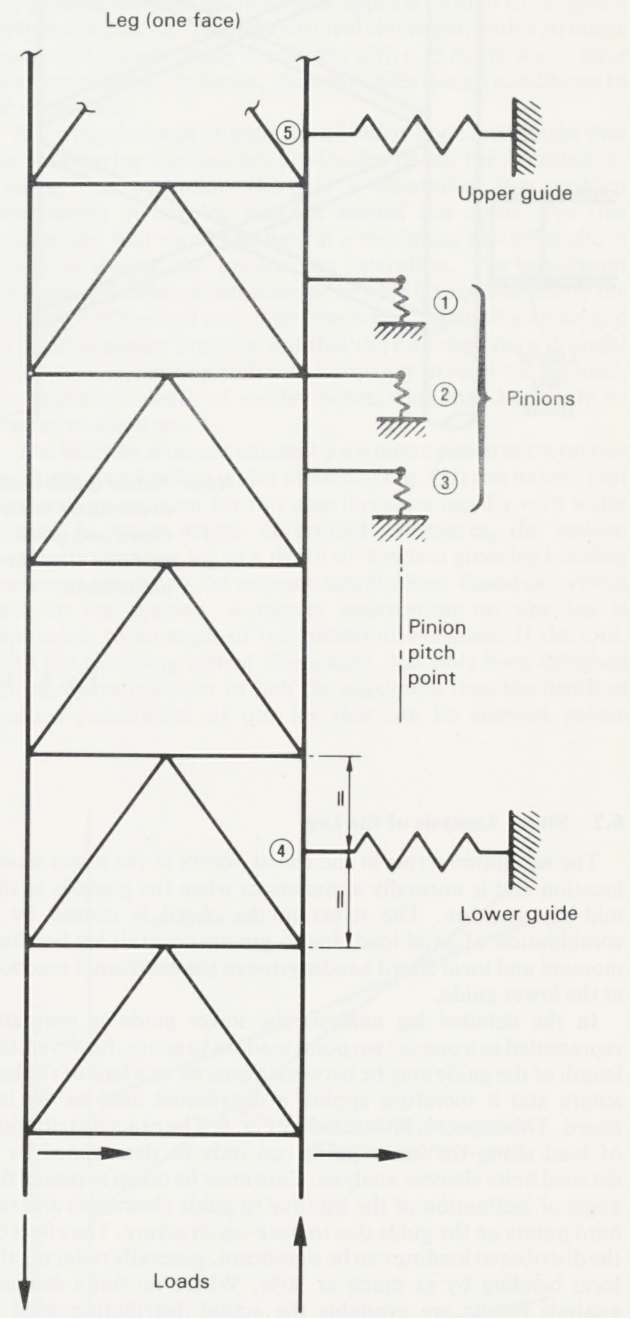

Fig. 5(b) Leg-pinion-guide idealisation

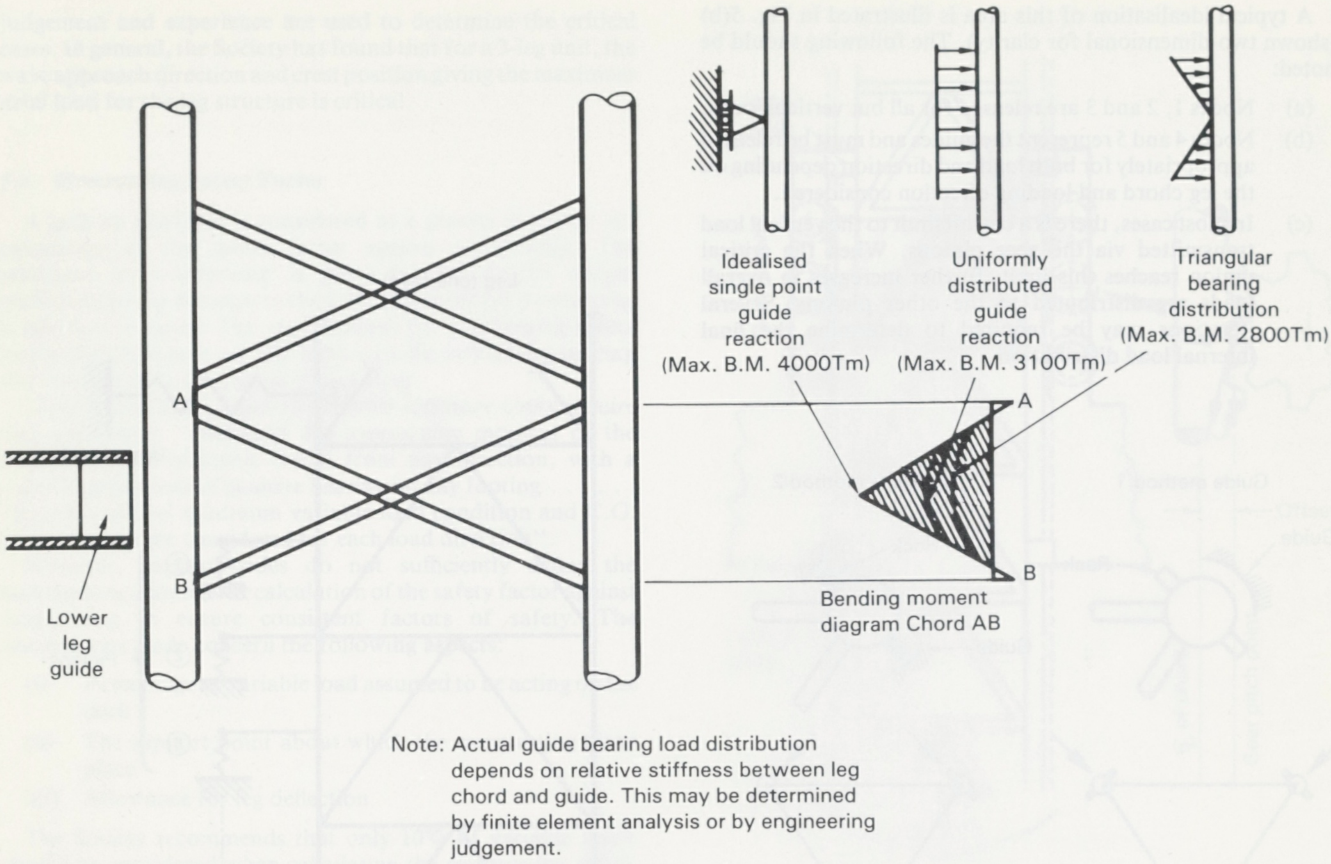

Fig. 6 Local chord bending

6.2 Stress Analysis of the Leg

The maximum stress in the chord occurs at the lower guide location and is normally a maximum when the guide is at the mid-bay position. The stress in the chord is caused by a combination of axial load due to gravity, overall leg bending moment and local chord bending due to the horizontal reaction at the lower guide.

In the detailed leg analysis the lower guide is normally represented as a one or two point load. In practice, however, the length of the guide may be between a quarter to a half of the bay length and it therefore applies a distributed load to the leg chord. This aspect is illustrated by Fig. 6. The exact distribution of load along the lower guide can only be determined by a detailed finite element analysis. Care must be taken to model the angle of inclination of the leg (due to guide clearance) and the hard points on the guide due to back-up structure. The effect of the distributed loading can be significant, generally reducing the local bending by as much as 30%. Where no finite element analysis results are available the actual distribution used is based on engineering judgement and consideration of the particular design feature. However, the double triangle distribution indicated on Fig. 6 has shown reasonable agreement with finite element results for some designs.

6.3 Allowable Stresses

The allowable stresses in both the chord and the braces are calculated according to the API RP 2A Code, section 2.5.[3] The allowables for punching shear at the joints are also given in this code.

7. LEG CLAMPING SYSTEMS

7.1 Purpose of Leg Clamping Systems

In the traditional design of a jack-up all the bending moment acting on the leg is reacted by the leg guides.

Many new jack-up designs are fitted with leg clamping devices. The purpose of the leg clamping device is to react the leg bending moment in differential vertical loads. This changes the way the leg is loaded and effectively reduces the loads in the leg in the following ways. Firstly, the maximum shear carried by the leg is reduced and hence the axial loads in the leg bracing are similarly reduced. This aspect is illustrated in the shear force diagram for a typical jack-up leg shown in Fig. 7. With all the bending moment reacted by the leg guides the shear in the leg between the guides is 3790 tonnes. When a clamping system is fitted and only 10% of the bending moment is taken by the guides, the shear is reduced by a similar proportion to 380 tonnes. With the shear in the leg reduced the loads in the braces are of course considerably reduced. Fig. 8 illustrates the change in stress for both diagonal and horizontal braces with change in the proportion of moment carried by the leg clamping system.

Secondly, the guide reactions are considerably reduced and consequently the local bending moment in the leg chord due to guide loading is also reduced. For example, in Fig. 7 the lower guide reaction is reduced from 4325 tonnes to 915 tonnes.

These reductions in leg loading naturally lead to a lighter leg design and may be essential for a deep water jack-up to be able to carry the full length of the leg when elevated during ocean tows. However, when the clamping system is not engaged the leg is effectively reduced in strength. It is therefore necessary to consider what loading cases may occur when the clamping system is not engaged.

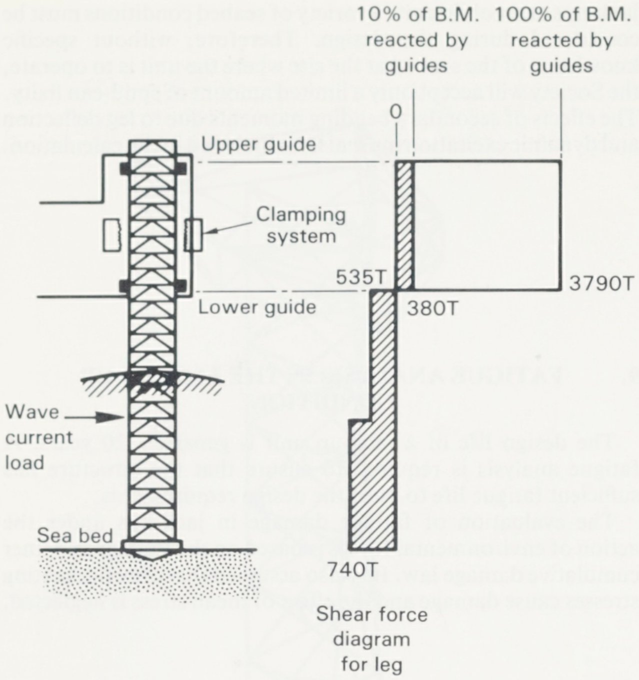

Fig. 7 Effect of % moment taken in clamping system (or pinions) on shear forces in leg

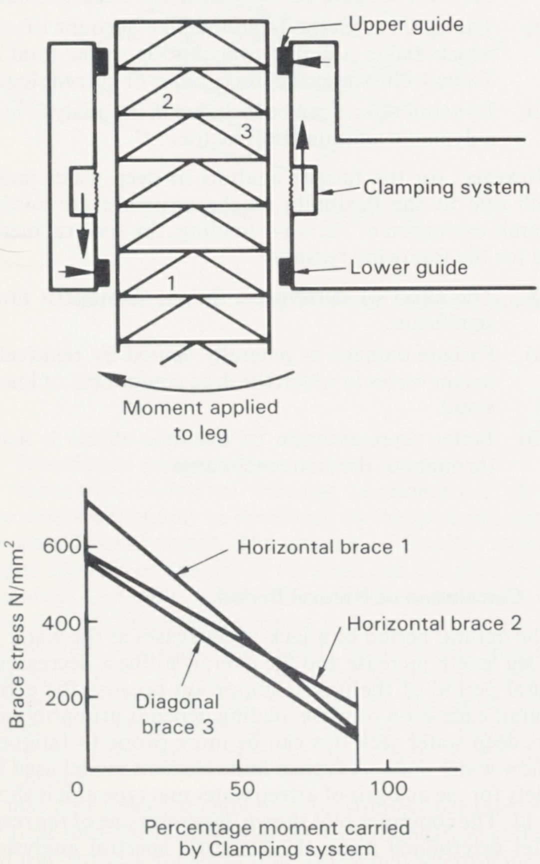

Fig. 8 Effect of % moment taken in clamping system (or pinions) on brace stress

7.2 Design Cases for Leg with the Clamping System not Engaged

The operating cases for jack-up units are not clearly defined. Some designers consider an operating wave of approximately half the extreme wave height to be a suitable design case for the jack-up leg with the clamping system disengaged and all the leg bending moment reacted by the leg guides. This will produce a leg bending moment of approximately 30% of the maximum survival condition. This is shown in Fig. 9 where the increase in leg bending moment against water depth is plotted for a typical deep water jack-up. Both the survival condition, with a 30 metre wave, and the operating condition, with a 15 metre wave, have been considered. However, there are other design conditions to be considered.

It is a requirement of both classification and certification that the jack-up leg footings are pre-loaded to test the foundation. During this operation there is a possibility that sudden penetration of one leg into the seabed can occur. For this reason, the hull should be held at a minimum elevation above sea level during the pre-loading operation. The maximum amount of sudden penetration of any leg is then limited by the buoyancy of the hull as it enters the water. Assuming an air gap of approximately 2 metres and that the hull requires a draught of 2 metres to develop sufficient buoyancy to react the leg load, the maximum depth of sudden penetration may hopefully be limited to 4 metres.

The bending moment caused by a 4 metre penetration on one leg during pre-loading is also plotted in Fig. 9. It can be seen that the bending moment for this case increases rapidly with water depth. At water depths of around 110 metres, the sudden penetration of one leg to a depth of 4 metres gives leg bending moments similar to the extreme storm cases. Based on typical jack-up leg spacing, 4 metres penetration on one leg is equivalent to an angle of inclination of 4 degrees. If the unit, with the clamping system disengaged, has only been designed for an operating wave of half the maximum then the depth of sudden penetration of one leg that can be allowed before

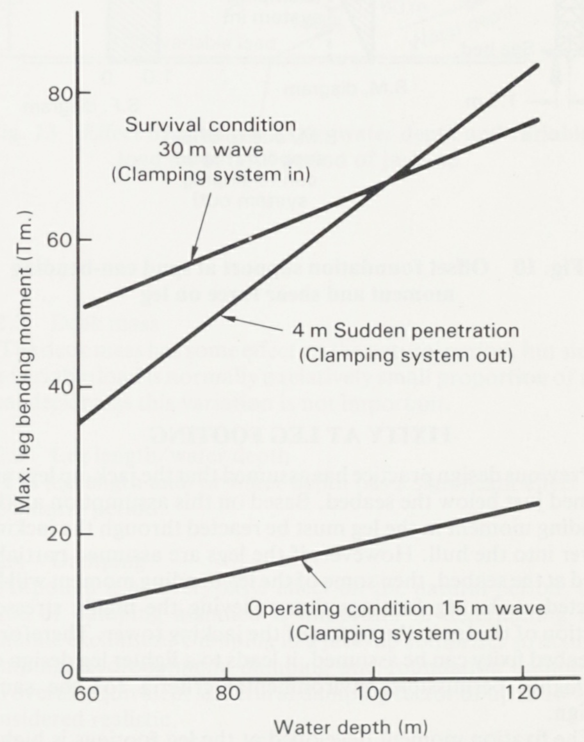

Fig. 9 Leg bending moment plotted against water depth for various load cases

reaching design stress levels is of the order of 1 metre. Again, based on typical jack-up leg spacing, this represents an angle of inclination of approximately 1 degree. That this angle of inclination should produce design stress levels is clearly unsatisfactory. In view of the above it is the author's conclusion that, for deep water jack-ups with leg clamping systems, the case of sudden leg penetration should be considered.

One other case that needs to be considered is that of offset foundation support at the spud-can while the unit is being jacked up. The effect of this case on a particular unit is shown in Fig. 10. The bending moment and shear force acting on the leg have been calculated for an offset of 1.5 metres. The extreme storm survival bending moment and shear force acting on the leg of the jack-up considered are also shown on the same diagram for comparison. The values of bending moment and shear force are shown as a ratio of the maximum extreme storm values. It can be seen that the maximum bending moment due to the extreme storm case is three times that for the offset support case. However, in the offset support case the clamping system cannot be engaged until the unit is fully jacked up. Therefore the bending moment is taken in the guides resulting in a shear force in the leg approximately 60% greater than that caused by the maximum storm wave. It is clear that the offset support case will effectively design the shear bracing in the leg.

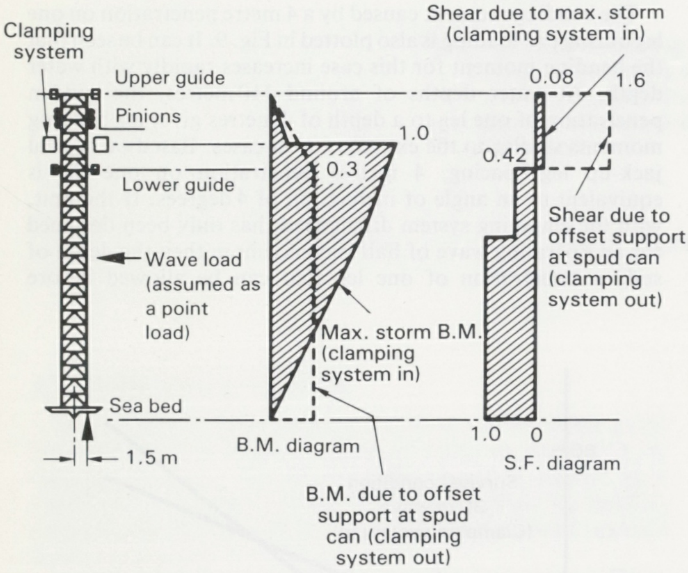

Fig. 10 Offset foundation support at spud can-bending moment and shear force on leg

8. FIXITY AT LEG FOOTING

Previous design practice has assumed that the jack-up legs are pinned just below the seabed. Based on this assumption all the bending moment in the leg must be reacted through the jacking tower into the hull. However, if the legs are assumed partially fixed at the seabed, then some of the leg bending moment will be reacted at the leg footings thus relieving the highly stressed portion of the leg in the region of the jacking tower. Therefore, if seabed fixity can be assumed, it leads to a lighter leg design or to higher permissible environmental criteria for the same design.

The fixation moment developed at the leg footings is highly dependent on the foundation and can only be taken fully into account where the soil conditions are exactly known and scour and other effects are very carefully monitored. However, since jack-ups are mobile units a variety of seabed conditions must be considered during the design. Therefore, without specific knowledge of the seabed at the site where the unit is to operate, the Society will accept only a limited amount of spud-can fixity. The effects of secondary bending moments due to leg deflection and dynamic excitation must also be included in the calculation.

9. FATIGUE ANALYSIS IN THE JACKED-UP CONDITION

The design life of a jack-up unit is generally 20 years. A fatigue analysis is required to ensure that the structure has sufficient fatigue life to meet the design requirements.

The evaluation of fatigue damage in jack-ups under the action of environmental forces is based on the Palmgren-Miner cumulative damage law. It is also assumed that only fluctuating stresses cause damage and the effect of mean stress is neglected.

9.1 Method of Analysis

For the strength analysis of drag dominated structures, such as jack-ups, the deterministic method is the most realistic and is to be preferred to the spectral method for the following reasons:

(i) Because of the fundamental assumptions made with regard to linearisation in the spectral analysis method, the wave loading can very easily be underestimated.

(ii) The spectral method cannot take account of current which makes a major contribution to the total loads. Table 1 illustrates the importance of current loading.

(iii) Dynamic effects can be included in the analysis by using a dynamic amplification factor.

However, for the fatigue analysis of deep water jack-ups, which due to the flexibility of the structure are subject to dynamic excitation of the wave loading, the spectral method is used for the following reasons:

(i) The effect of current on fatigue damage is not very significant.

(ii) Fatigue damage is generally caused by relatively low height waves in which the drag component of loading is small.

(iii) Better representation of dynamic effects is achieved throughout the frequency range.

9.2 Calculation of Natural Period

The natural period of a jack-up increases as the water depth and leg length increase and the overall stiffness decreases. The natural period of the unit is important because the extent of dynamic excitation of wave loading depends primarily on this. Thus deep water jack-ups can be more prone to fatigue than shallow water units. A typical finite element model used by the Society for the analysis of a deep water mat type unit is shown in Fig. 11. The computer plot shown illustrates one of the response modes determined using the Society's spectral analysis program.[4] The deflections are exaggerated.

Very careful consideration must be given to the assumptions made in determining the natural period. Some of these assumptions are discussed in the following sections.

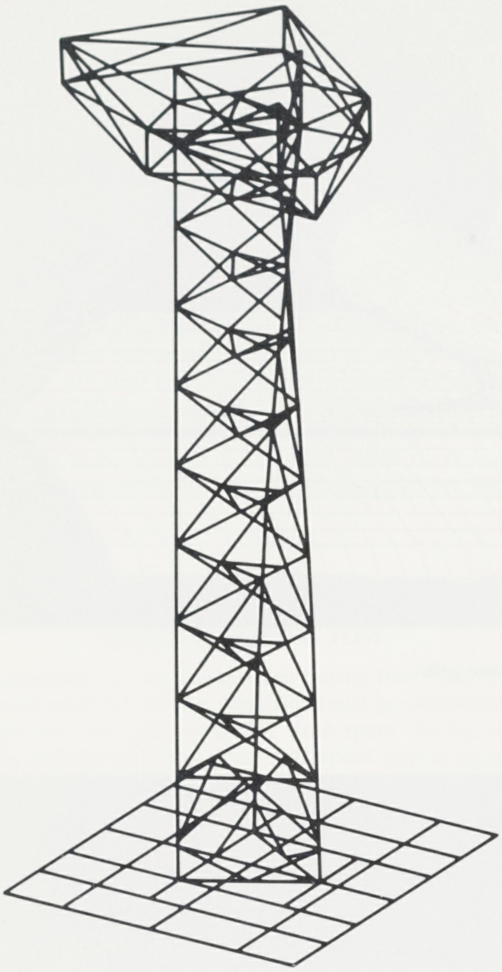

Fig. 11 Deflected plot showing torsional response mode

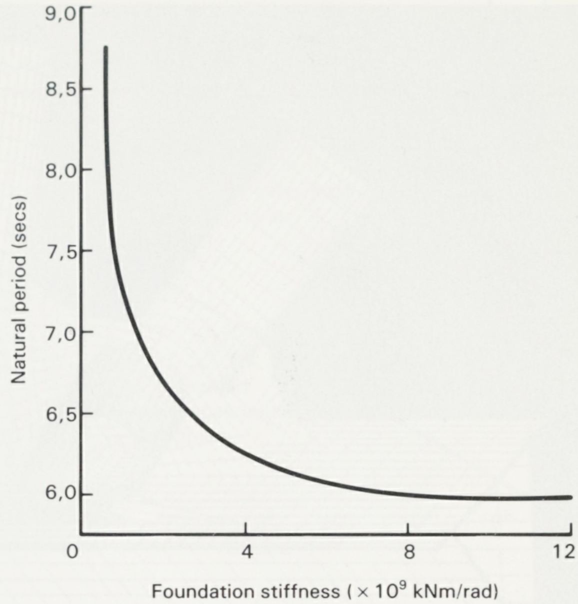

Fig. 12 Variation in natural period with foundation stiffness for a deep water mat supported jack-up

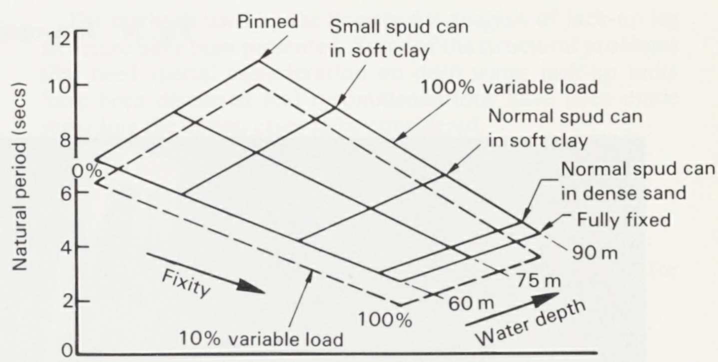

Fig. 13 Effect of spud can fixity, water depth and variable load on natural period of jack-up

9.2.1 Foundation assumptions

For units with individual footings consideration of the fixation moment that may be developed by the spud can resting on the seabed is important. The amount of fixation moment developed depends on the size of the spud can and the stiffness of the seabed. Generally, unless soil conditions are known, it is assumed that the foundation stiffness is the minimum that can occur consistent with achieving a safety factor on the spud can bearing capacity of 1.0 under the pre-load condition. Based on this foundation stiffness and the size of the spud cans, rotational, horizontal and vertical springs are selected to represent the spud can/foundation interaction in the finite element model.

For mat type units the choice of soil stiffness parameters is also critical since the relationship between foundation stiffness and natural period is not linear (Fig. 12). When determining the soil stiffness an average value may be assumed taking account of the increasing stiffness with depth. Typically the Society has considered a value equal to the average soil stiffness over a depth equivalent to $\frac{1}{3}$ of the mat width.

9.2.2 Deck mass

The deck mass has some effect on the natural period, but since the variable load is normally a relatively small proportion of the total deck mass this variation is not important.

9.2.3 Leg length/water depth

The leg length and/or water depth has a significant effect on the natural period.

9.2.4 Damping

Although it has very little effect on the natural period, the value of damping assumed is important in determining the dynamic excitation. Damping in a jack-up comprises structural damping, hydrodynamic damping and foundation damping. An overall equivalent structural damping factor of up to 10% is considered realistic.

The combined effect of the above assumptions for a particular unit can be shown as a carpet graph (see Fig. 13). It can be seen that a very wide range of natural frequencies can be obtained depending upon the assumption made.

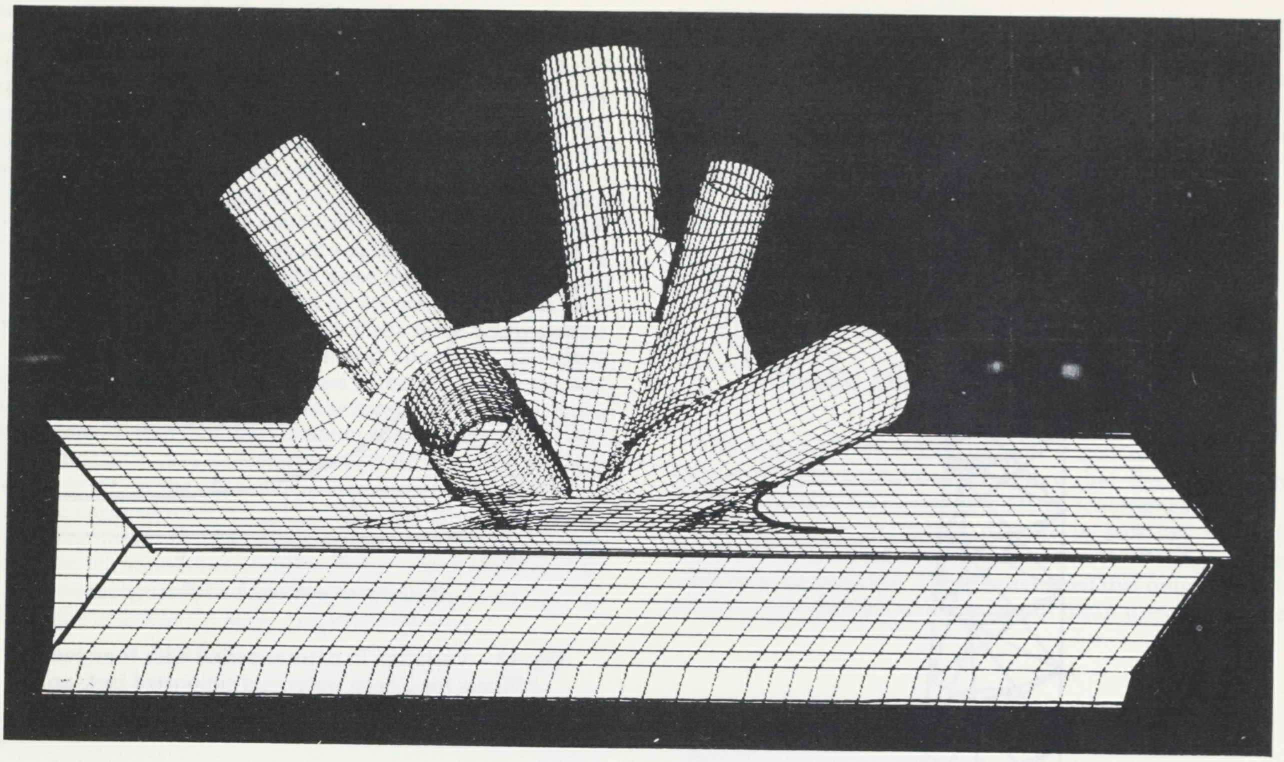

Fig. 14 Jack-up chord/brace joint

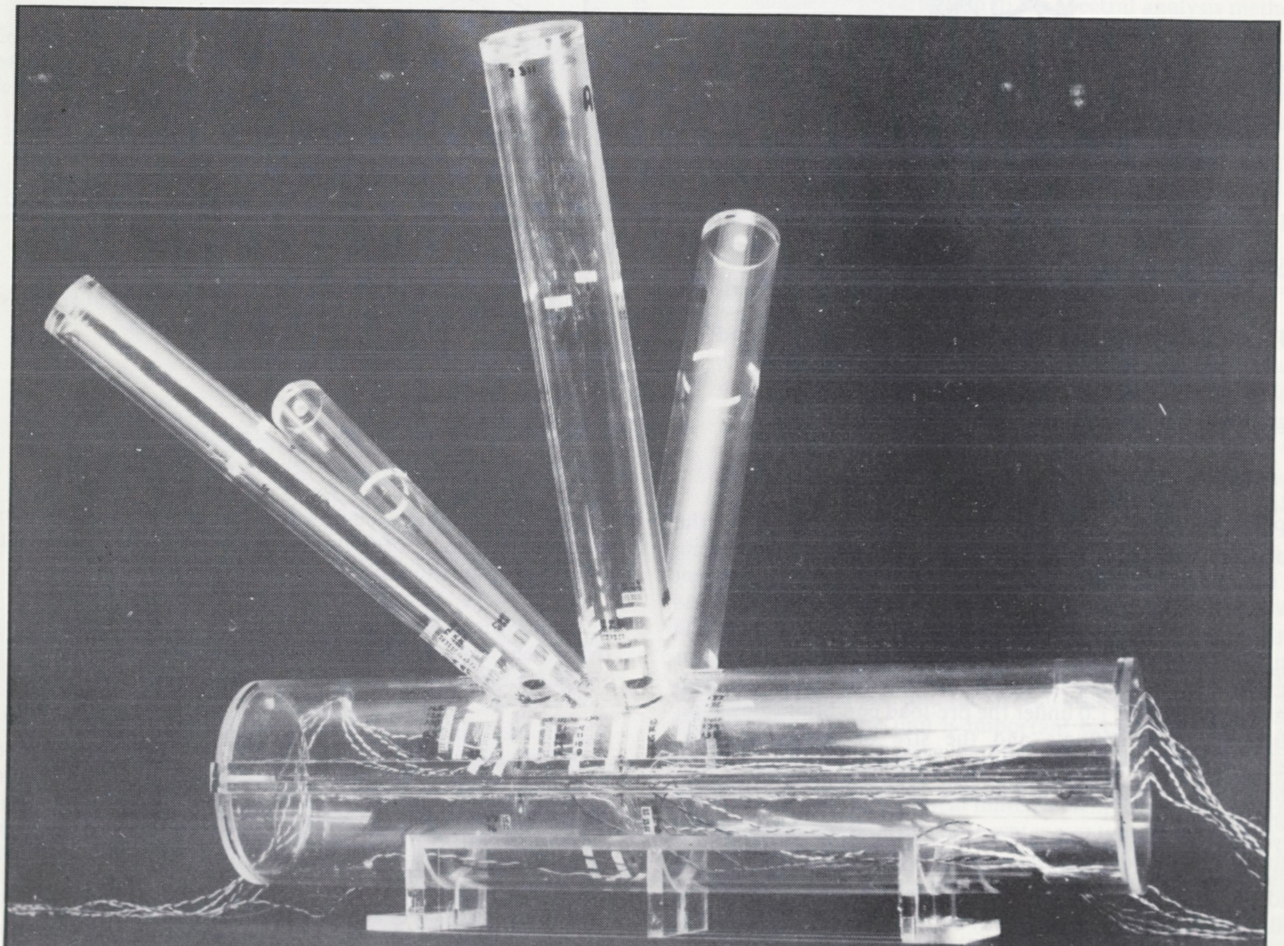

Fig. 15 Acrylic model of jack-up chord/brace joint

Assuming a minimum soil stiffness based on pre-load requirements and 50% variable deck load a natural period of the order of 7.0 seconds is obtained. This compares with a maximum value of 11.0 seconds for the leg assumed pinned at the seabed and 100% variable load.

This value of natural period is still considered conservative for fatigue calculations as it assumes maximum water depth for the whole life of the unit.

9.3 Stress Concentration Factors

An important parameter for fatigue analysis is the stress concentration factor assumed at the jack-up leg connections. The application of parametric formulae developed for tubular connections on fixed steel platforms is not always possible since jack-up leg connections vary significantly in design. Often the main chord member is not tubular and, even when it is a tubular section, there are frequently through plates that restrict chord ovalisation and also external gussets. For this reason it is sometimes necessary to carry out an investigation of the particular joint or joints. This can be done using either acrylic modelling or finite element techniques. Both these methods have been employed by the Society and examples are shown in Figs. 14 and 15.

10. FATIGUE DURING TOW

The possibility of fatigue damage during tow when the legs are fully elevated for the tow condition must be considered. For a jack-up unit the proportion of time spent during tows is generally considerably less than that spent operating at site. However, unlike the jacked-up case, during tow the high stress point on the leg is always at the same position. Two locations in particular on the leg need to be considered. Firstly, the leg joints in the region of the upper guide and in the jacking tower generally. At this location the maximum overall bending moment and local bending moment due to the guide reaction occur. Maximum shear also occurs in this region. Secondly, where a leg clamping system is fitted and is used during the tow, the rack teeth on the leg at this location need to be specially considered. Detailed discussion of the leg clamping system is not possible since the clamping system is specific to each design. In general, however, welded connections should be avoided in the section of the leg rack engaged with the clamping system during tow. This section may also require machining to ensure that the tooth loading is as uniform as possible for both directions of loading.

For the fatigue analysis during the tow maximum stresses can be determined from the motions as established by model tests. The number of stress cycles are determined from the anticipated maximum time spent on tow during the life of the unit and the average period of the motions expected. The main uncertainty in the calculation is the shape of the stress/exceedance spectrum. At present the Society assumes a straight line relationship between stress plotted on a linear scale and cycles plotted on a log scale (see Fig. 16). Further data on this aspect is required before more exact fatigue calculations can be performed.

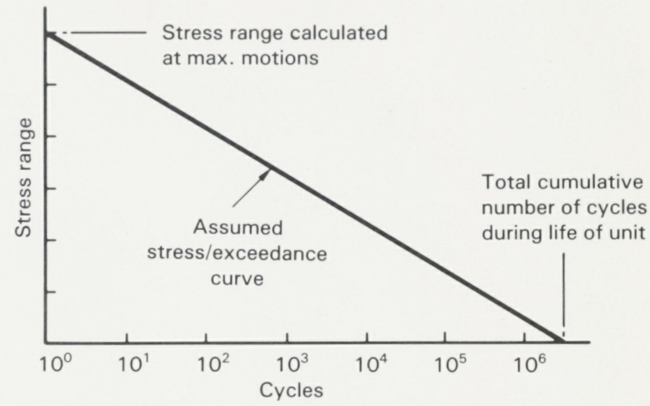

Fig. 16 Assumed stress/exceedance curve for tow

11. CONCLUSIONS

The methods used by the Society for analysis of jack-up leg structure have been presented. Some of the structural problems that need special consideration on deep water jack-up units have been discussed and recommendations have been made regarding the design cases to be considered.

ACKNOWLEDGEMENT

The author is indebted to his colleagues in the Society for their help in the preparation of this paper.

REFERENCES

1. Rules and Regulations for the Classification of Mobile Offshore Units, 1984; Lloyd's Register of Shipping.
2. S. Singh *et al*: "Wave Forces on Circular Cylinders with Large Excrescences at Low Keulagan and Carpenter Numbers", NMI Report R. 133, March 1982
3. American Petroleum Institute. Recommended Practice for Planning, Designing and Constructing Fixed Offshore Platforms, API RP 2A.
4. Bainbridge C. A. and Nataraja R., Lloyd's Register of Shipping; Dynamic Analysis of Fixed Platforms. "Offshore China '83" Conference.

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Lloyd's Register Technical Association

Discussion

on the paper

STRESS ANALYSIS OF SELF-ELEVATING PLATFORMS

by

W. J. Winkworth

Paper No. 5. Session 1986–87

FOR PRIVATE CIRCULATION AMONGST THE STAFF ONLY

Any opinions expressed and statements made in this Discussion Paper are those of the individuals.

Hon. Sec. R. V. Pomeroy
71 Fenchurch Street, London, EC3M 4BS

Discussion on the Paper

STRESS ANALYSIS OF SELF-ELEVATING PLATFORMS

by

W. J. Winkworth

DISCUSSION

From Mr. P. J. Fisher:

I would first like to congratulate Mr. Winkworth on a most interesting and informative paper. It provides an extremely useful document including comprehensive coverage of the Society's approach to the structural appraisal of jack-up legs.

As a result of the current considerably reduced oil price, there has been a large reduction in offshore exploration leading to many jack-up units being laid up. At present, the oil companies are searching for more economical methods to exploit offshore oil and gas. My question is related to the fact that many oil companies are now looking at jack-up units as economical production and accommodation platforms.

Jack-ups are generally employed as drilling units operating in different water depths for relatively short periods. Using jack-ups as production or accommodation platforms would necessitate them operating at a constant water depth for long periods. I would like to ask Mr. Winkworth what additional requirements need to be considered when jack-up units are used in this way?

From Mr. R. B. Siggers:

The author is to be complimented for a very useful paper, well presented. It is understood that the paper was originally proposed as a joint OED/MDA paper so hopefully the following notes may be found to be of interest.

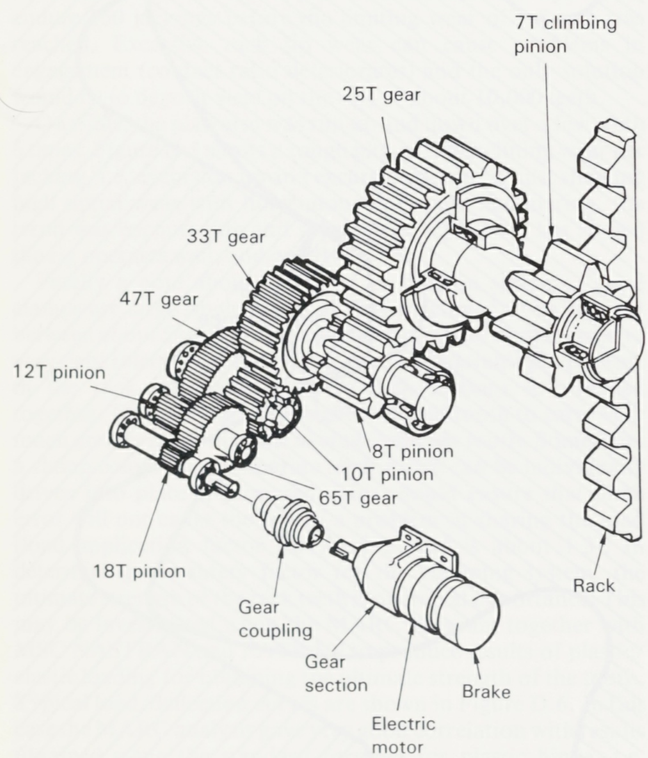


Fig. D.1 Gear Train for 700 Series Jacking System

Engineering input from MDA in jack-up appraisal has grown over recent years and one of these contributions is in gearing (on gear driven self-elevating platforms), although work is also carried out on hydraulic actuator driven systems.

Gears are normally precision made components working in a reasonably protected environment. This is not the case for the rack and pinion gears used for elevating platforms. Figure D.1 shows a typical gear train for one rack and pinion engagement.

Design Calculations for strength assessment follow the normal procedures adopted for shafts, bolts, keys, welds etc. in torque transmission systems.

Similarly the gearing between prime mover and final rack and pinion is examined for tooth strength throughout the gear train to Rules (for Hertz and Bending stresses).

However, when the final rack and pinion drive is reached, a whole set of new problems arise basically because of the essential differences of the system. Some of these are:-

(i) Accuracy
(ii) Stress levels/loadings—shakedown limits
(iii) Friction factor/lubrication
(iv) Geometry: Circular arc or involute
(v) Varying meshing positions—floating engagement
(vi) Manufacturing processes and size.

(i) *Accuracy*. The accuracy of a marine propulsion gear is dependent on the machining, i.e. hobbing or planing followed by post-hobbing processes and tooth corrections for longitudinal deflections arising from bending and twisting etc. and accuracy is assessed by measuring profile errors, pitch errors, radial run out, roughness etc. The size of these values is small, i.e. in microns (micro-metre). On a jack-up pinion, radial run out (eccentricity) is measured in millimetres and can be as much as 4 mm on flame cut pinions.

(ii) *Stress Levels*. The stresses permissible for a rack and pinion gear will be much higher than for marine propulsion. The gear will have a design life measured in hours *not* infinite life and the speed of the transmission is very low—about half a metre per minute.

The limiting Hertz stress is governed by the shakedown limit. (Shakedown is the name given to the process of inducing residual stresses by repeated cycles of loading, usually less than 6 cycles). The plastic deformation which occurs during the shakedown cycles ceases and the load is then supported by elastic stresses. However at very high loading this stable condition will not occur and progressive deformation will take place until the teeth are severely damaged. The upper limit for Hertz stress permitted by the Society for normal operation is 3 times the yield stress of the weakest material (usually the rack). Emergency (low cycles) operation at higher loadings is often specified and, for this level of stress, the gears may suffer damage for which acceptance by the operator is sought. The limiting Hertz stress for this condition is 3.5 times yield stress. It must be remembered that the weakest link in the whole system from prime mover through to the leg of the platform is often specified to be the rack teeth.

Additional evaluation of stress is by assessment of ultimate strength (plastic hinge) criterion, i.e. calculation of the collapse load which will produce yield through the full thickness of the tooth. Again, the rack is to have the lowest collapse load and it should have a factor of safety of not less than 1.1 on the maximum storm survival load specified.

(iii) *The Friction Factor*. The rack and pinion loading used on strength assessment is necessarily that which the teeth will experience. When lifting the platform this will be increased because of friction forces from the teeth and also from the rack guide system. Normally this factor is set at 10%, so the lifting load is 110% and the lowering

load 90% of the platform weight. Other factors to be considered are the load variation of each pinion on each rack, on each chord (i.e. across 2 racks) and on each leg; the brake slipping loads; the effect of variation in location of the centre of gravity of the platform; and possible load reversals which can occur during an ocean tow. In order to reduce friction levels to a minimum, expensive grease and greasing systems are used to reduce not only friction forces but also noise and vibration. At very high rolling and sliding loads lubrication of any kind is often temporarily suspended and adhesive wear takes place at exceedingly high noise levels.

(iv) *Tooth Geometry*. Often gear tooth profiles are not of standard shape (which is involute), but can be part involute and part circular arc, or even all circular arcs. The meshing geometry can produce transverse contact ratios of less than unity but although pinion tooth deflection will improve this, the Society expects design values to be over unity without deflection.

(v) *Varying Meshing Position*. On account of the errors and tolerances inherent in rig construction, the rack and pinion engagement can have a maximum range of movement of the rack of about 25 mm, and this has to be taken into consideration in calculating contact ratio, root bending stresses, and collapse loads. Further examination has to be made by computer simulation to establish that there is no tight mesh, undermesh or root and tip interference during engagement (Figure D.2) under the worst conditions of fabrication tolerances of the chord and rack straightness; rack teeth and pinion teeth pitch, profile errors and radial run out (see Figure D.3). A computer simulation should reveal any possible problems of this nature.

(vi) *Manufacturing Process*. The racks are normally flame cut from slabs of high yield (about 700 N/mm^2) weld quality steel in lengths of about 20 teeth and welded onto the chord. The pinions are either of cast or forged construction and often flame cut and hand dressed by grinding. A common pinion would have 7 teeth, 110 mm normal module with a tooth height of 250 mm and a yield strength about 30% higher than that of the rack.

A typical jack-up having 3 legs, with three chords per leg, and 2 racks per chord, would require about 10,000 rack teeth and about 50 pinions.

When a jacking system is submitted for approval, we look at the aggregation of the worst possible conditions and see how the gear engagement and loadings will behave. From the foregoing it can be seen that the control of the engagement depends on many tolerances but ultimately the leg chords (racks) must be constrained between the guide distance. Recent experience has shown that the problem of guide wear cannot be neglected. The predicted lateral forces on the guides are known, and by making certain assumptions some calculations can be made regarding wear rates.

On a large jack-up, high bending moments can occur on the legs during jacking. These bending moments cause high lateral loads on the leg guides and on the rack tips where the leg is guided on the rack. A recent case is illustrated in Figure D.4. Because of guide clearances, the leg is angled to the guide plate and very high loads occur on one tooth over a short distance and this leads to wear on both the rack teeth and guide plates. Background information on adhesive wear under pressures is

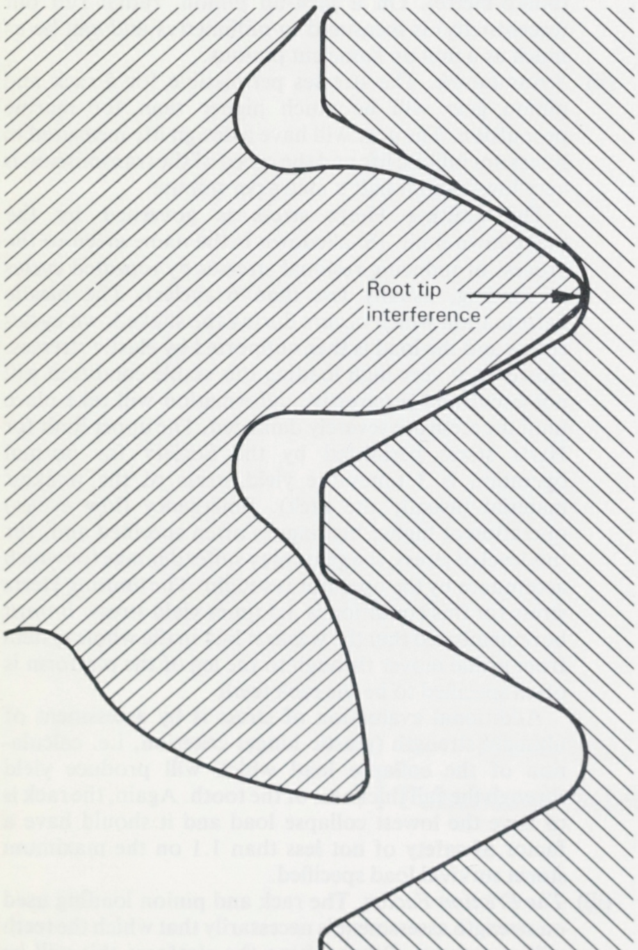


Fig. D.2a

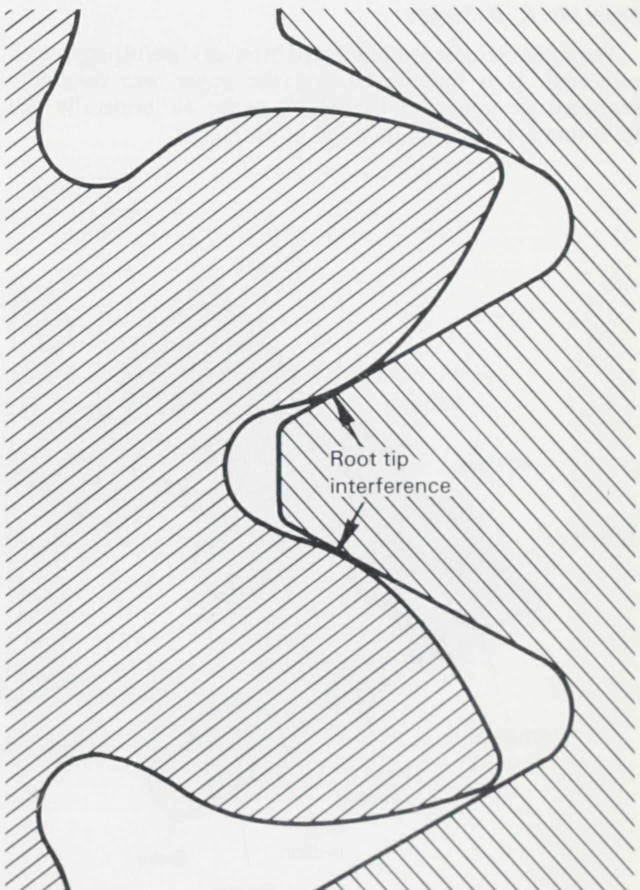


Fig. D.2b

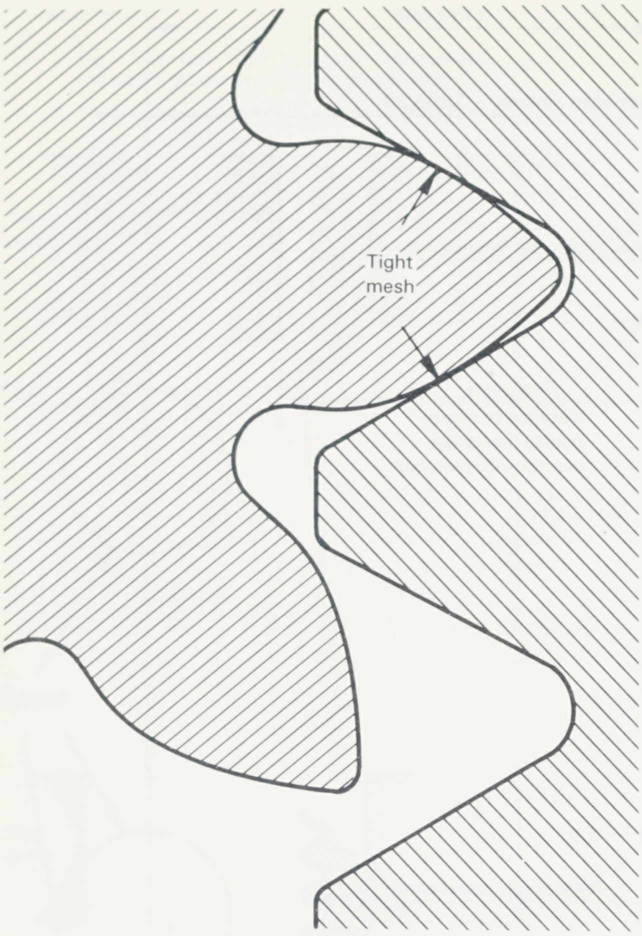


Fig. D.2c

not readily available but we predicted that the rack tips would endure 150 jackings before the limiting wear of 1.8 mm was reached. Excessive rack tip wear can cause problems in engagement (contact ratio deteriorates) and the only solution would be to deposit weld on the tips of about 10,000 teeth.

On trials, the platform was run up and down over a few teeth 6 times. Figure D.5 shows a rough plot of the declining wear per jacking, i.e. excursion up and excursion down. Whilst showing high initial wear with horrendous noise and hammering, the trend was encouraging and subsequent experience has shown the rig operates well, and quietly.

Finally a note about clamping or fixation systems. Some clamps are large single pieces of steel machined to be inserted between about 5 teeth. Although the clamp may be machined to very tight tolerances, there will still be considerable pitch errors in the rack, particularly where rack sections are welded together. This will cause the highest loaded tooth to carry very much greater loads than the average load per tooth. Sometimes a clamp comprising 5 separate teeth which can be individually driven into place is proposed. This should ensure that pitch error will not cause too severe a problem in sharing the load (load application factor for sharing load is about 1.3). In determining the safety factor for the clamping system, the ultimate strength of the rack teeth is of crucial importance. This may be investigated using the MARC program together with MSC NASTRAN and PATRAN to produce results of plastic/elastic loading for obtaining the ultimate strength of the tooth. Typical load/deflection curves are shown in Figure D.6. In this case the MARC analysis gave very good correlation with results obtained using the standard formula for plastic hinge, i.e. applying the platform load via the clamp tooth to the rack at the pitch line gave a collapse load of 1800 tons on the rack tooth.

It is hoped that the above brief description gives some idea of the work undertaken by Machinery Design Appraisal Department when reviewing design of gear driven self-elevating platforms.

From Mr. G. B. Singh:

The author is to be congratulated for producing a very concise and interesting paper on the strength analysis of self-elevating units.

The paper covers different aspects of the analysis of jack-up platforms in a systematic way, which are not easily found without much reading and effort. Therefore, I am sure this paper can be of great assistance to structural engineers in understanding the fundamentals of the structural analysis and design of a jack-up, particularly the leg structure.

However, would the author please comment on the following structural aspects of design appraisal which are not covered by the paper:

1. It appears that any damage to a leg of a deep water jack-up (particularly a 3-leg structure) due to boat impact could seriously impair the structural integrity of the platform. Has any boat damage assessment been carried out by the Society, and what are the current classification/certification requirements regarding the design for such loading conditions?

   If presently, there are no specific requirements/guidance for boat impact, then what is the current D.En. thinking (i.e. is there any draft guidance) regarding this aspect of design?
2. Jack-up platforms have now been in service for a long time. Consequently some will have suffered damage during their operating life. What sort of damage has been reported and what were the consequences of the damage? Has the experience gained from any such damage helped to improve the current method of analysis/design?

From Mr. R. V. Stanford:

For self-elevating units, the Classification Societies recommend different criteria to assess the leg strength in transit conditions. Can the Author please explain the background to the method used by the Society to assess the leg bending moments in transit conditions as it would appear that the Society's requirements are more severe than those used by other Classification Societies?

From Mr. C. S. Whitcroft:

I would like to thank Mr. W. Winkworth for a most interesting and stimulating paper.

Recognising the practical problems of rewelding high tensile grade leg members *in situ,* and the high costs involved, has the Society been asked to consider mechanical systems for reconnecting leg sections; would such techniques be viable?

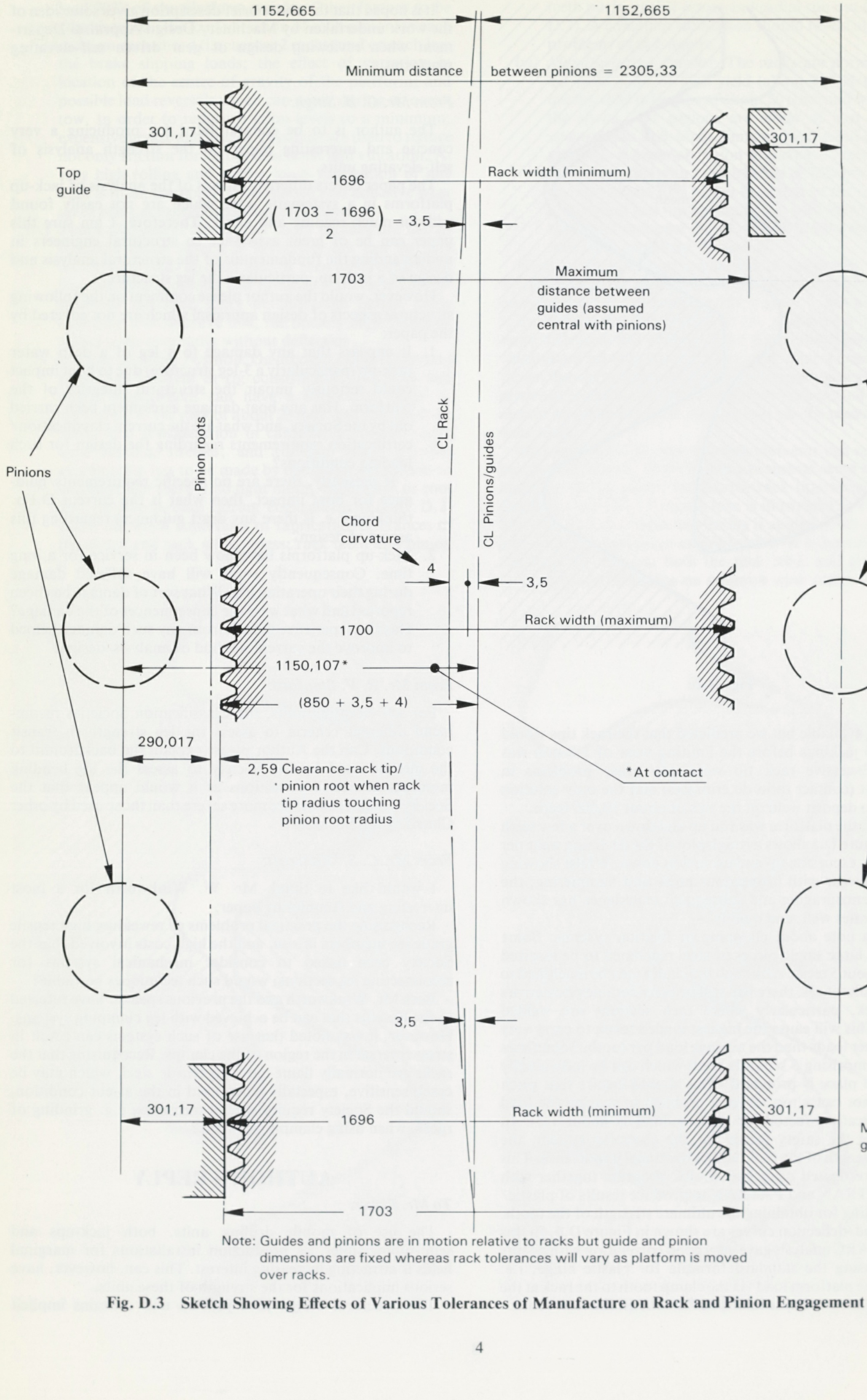
Both Mr. Winkworth and the previous speaker have referred to the benefits that can be achieved with leg clamping systems. However, it was noted that use of such systems can result in stress reversal in the region of the clamps. Recognising that the racks are normally flame cut high tensile steel, which may be crack sensitive, especially when used in the as-cut condition, should the Society require special measures, e.g. grinding of racks, when using clamping systems?

AUTHOR'S REPLY

To Mr. Fisher:

The use of mobile drilling units, both jack-ups and semi-submersibles, as production installations for marginal fields is attracting increasing interest. This can, however, have serious implications for the strength of these units.

The structural design of a jack-up unit contains implicit



Note: Guides and pinions are in motion relative to racks but guide and pinion dimensions are fixed whereas rack tolerances will vary as platform is moved over racks.

Fig. D.3 Sketch Showing Effects of Various Tolerances of Manufacture on Rack and Pinion Engagement

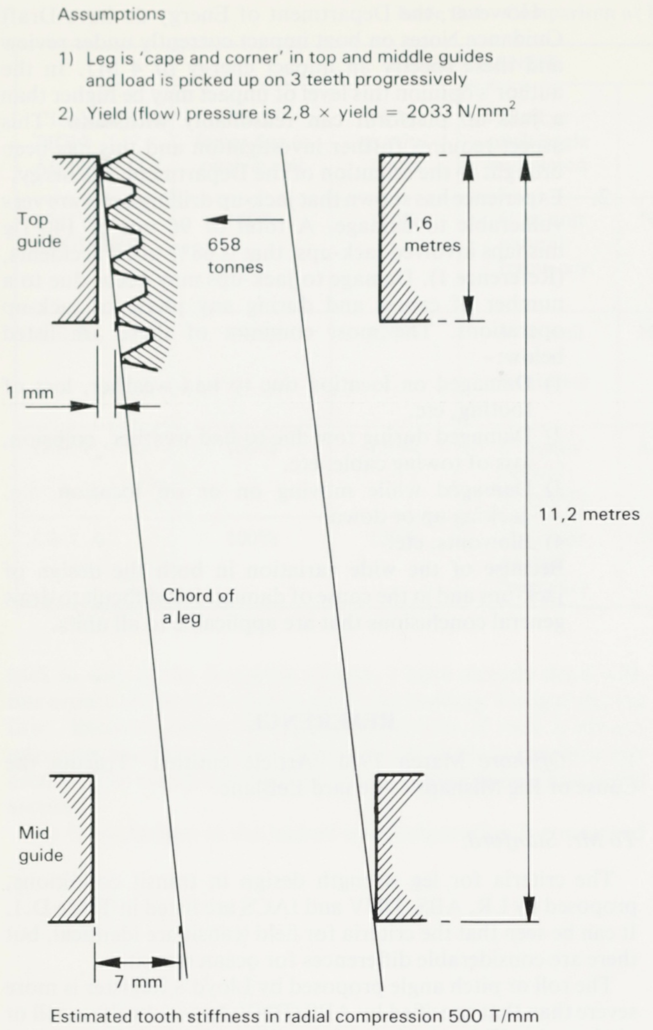


Fig. D.4 Basic Assessment of Rack Tip Wear Rate

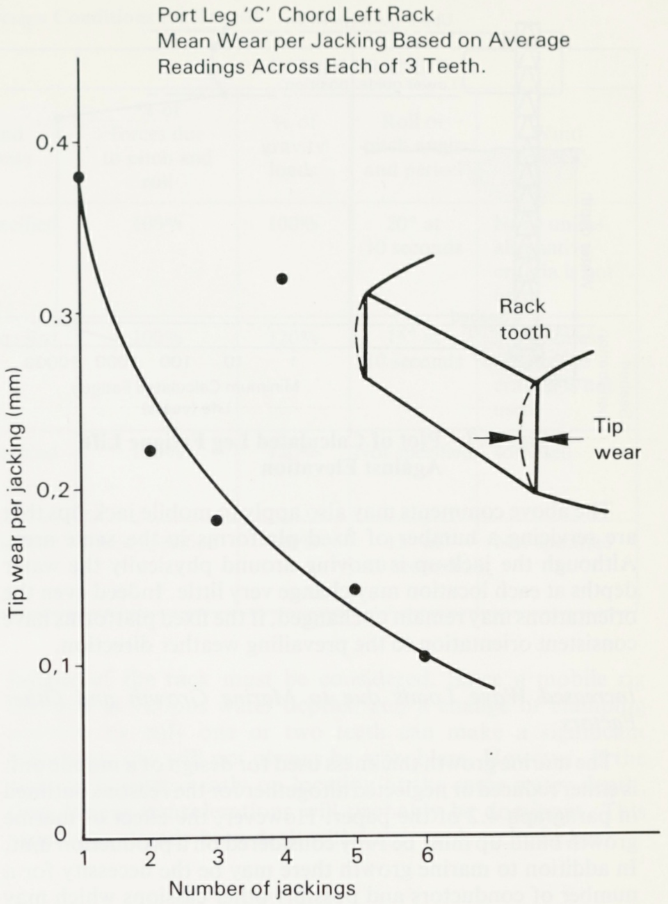


Fig. D.5

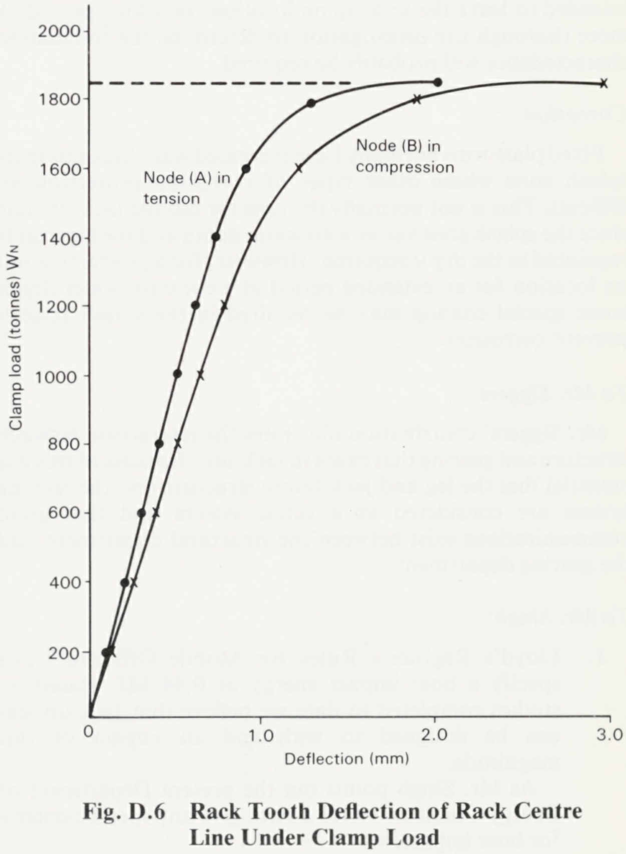


Fig. D.6 Rack Tooth Deflection of Rack Centre Line Under Clamp Load

assumptions about how the unit will eventually be operated. Therefore a structural re-analysis of the design should be undertaken whenever the unit is to be used for a type of operation or environmental criteria for which it was not designed. This applies particularly where the unit is used as a production installation since it will generally be fixed at one location and this represents a considerable change in the method of operating. Some of the structural design aspects to be considered in this case are discussed below.

Fatigue

Typically, jack-ups are designed for water depths varying from 100 ft or less to 350 ft. For a mobile unit the elevation of the barge with respect to the leg changes with varying water depths and seabed penetrations. From Figure D.7 it can be seen that the fatigue lives of the brace connections in way of the lower horizontal guide and jack-house are considerably lower than other parts of the leg. Even at one bay length below the lower guide location there is a considerable increase in fatigue life. For a mobile jack-up operating in areas of varying water depths the period at any one site may not be more than a few months. Hence the location of the maximum fatigue damage is spread over 150 ft or more of the leg, instead of being concentrated at one position. This effectively increases the fatigue life of the leg. For a production jack-up, fixed at one location, this advantage is lost and therefore the fatigue life of the critically loaded part of the leg will be very much reduced.

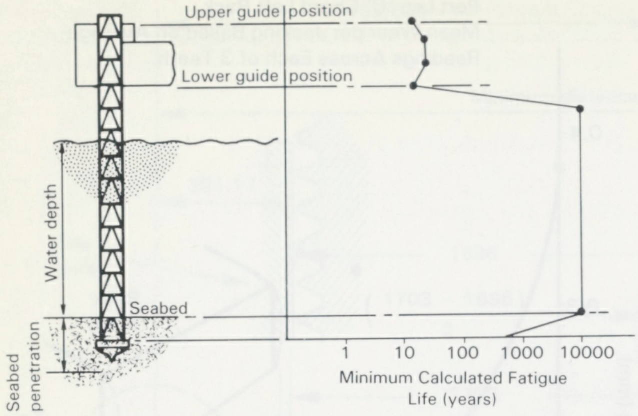


Fig. D.7 Plot of Calculated Leg Fatigue Life Against Elevation

The above comments may also apply to mobile jack-ups that are servicing a number of fixed platforms in the same area. Although the jack-up is moving around physically the water depths at each location may change very little. Indeed even the orientations may remain unchanged, if the fixed platforms have consistent orientation to the prevailing weather direction.

Increased Wave Loads due to Marine Growth and Other Factors

The marine growth thickness used for design of a mobile unit is either reduced or neglected altogether for the reasons outlined in paragraph 4.2 of the paper. However, the effect of marine growth build-up must be fully considered on a production unit. In addition to marine growth there may be the necessity for a number of conductors and possibly other cassions which may significantly increase the wave loading.

Foundations

The foundation must be specially considered where it is intended to leave the jack-up on location for a long period. A more thorough site investigation to determine the foundation characteristics will probably be required.

Corrosion

Fixed platforms normally have increased wall thickness in the splash zone where other types of corrosion protection are difficult. This is not normally the case for mobile jack-up units since the splash zone varies with water depth and the legs can be repainted in the dry if required. However, for a production unit on location for an extended period at a constant water depth, some special coating may be required in the splash zone to prevent corrosion.

To Mr. Siggers:

Mr. Siggers' contribution illustrates the interaction between structure and gearing that exists in jack-ups. Because of this it is essential that the leg and jack-house structure and the jacking system are considered as a single system and that good communications exist between the structural department and the gearing department.

To Mr. Singh:

1. Lloyd's Register's Rules for Mobile Offshore Units specify a boat impact energy of 0.44 MJ. Based on studies completed to date we believe that jack-up legs can be designed to withstand an impact of this magnitude.

   As Mr. Singh points out the present Department of Energy Guidance Notes do not give any specific criteria for boat impact.

   However, the Department of Energy do have Draft Guidance Notes on boat impact currently under review and these specify an impact energy of 4 MJ. In the author's opinion this level of impact may be higher than a jack-up platform can reasonably withstand. This aspect requires further investigation and this has been brought to the attention of the Department of Energy.

2. Experience has shown that jack-up drilling units are very vulnerable to damage. A total of 96 out of 140 rig mishaps involved jack-ups, that is 68% of all accidents, (Reference 1). Damage to jack-ups may occur due to a number of causes and during any phase of jack-up operations. The most common of these are listed below:-
   1) Damaged on location due to bad weather, loss of footing, etc.
   2) Damaged during tow due to bad weather, collision, loss of towing cable, etc.
   3) Damaged while moving on or off location, i.e. jacking up or down.
   4) Blowouts, etc.

   Because of the wide variation in both the design of jack-ups and in the cause of damage it is difficult to draw general conclusions that are applicable to all units.

REFERENCE

1. *Offshore* March 1981. Article entitled 'Tracing the Cause of Rig Mishaps'—Lenard LeBlanc.

To Mr. Stanford:

The criteria for leg strength design in transit conditions, proposed by LR, ABS, DNV and IACS are listed in Table D.1. It can be seen that the criteria for field transit are identical, but there are considerable differences for ocean transit.

The roll or pitch angle proposed by Lloyd's Register is more severe than that specified by ABS (DNV do not specify a roll or pitch angle). This criteria has been chosen by the Society based on reports of actual tows which indicated that accelerations can occur which are greater than those given by $\pm 15°$ at 10 seconds.

In our experience this has not created a problem with clients to date because:-

(i) The Society allows the use of motions based on model tests and/or calculations and this alternative is often adopted.

(ii) Many of the organisations offering warranty survey for insurance purposes specify criteria that is very similar to Lloyd's Register. Therefore the designers have to meet this criteria regardless of class requirements.

To Mr. Whitcroft:

The possibility of connecting leg sections by mechanical means has been discussed but no firm proposals have been submitted to the Society for approval as far as the author is aware. Any mechanical connection device would be of enormous benefit because it would enable a rig to be towed with relatively short legs and to extend the legs when it arrived on location. This can of course be done by welding but this is a very expensive operation, particularly at exposed or isolated locations. However, there are considerable technical difficulties to be overcome with a mechanical leg connection. The leg chord has to pass through the leg guides and jacking system, so no outstanding sections such as flanges etc. can be allowed. In addition, very high stresses occur in the leg chord, particularly at the bay in way of the lower guide. Because leg penetrations vary it would not normally be possible to ensure that the bay with the mechanically connected chord would not be in this position.

With regard to fatigue in both the clamping system and the

Table D.1 Comparison of Leg Design Conditions for Transit

CLASSIFICATION AUTHORITY	FIELD TRANSIT				OCEAN TRANSIT			
	% of forces due to pitch & roll	% of gravity loads	Roll or pitch angle and period	Wind velocity	% of forces due to pitch and roll	% of gravity loads	Roll or pitch angle and period	Wind velocity
LLOYD'S REGISTER	100%	120%	6° at unit natural frequency	Not specified	100%	100%	20° at 10 seconds	None unless alternative criteria is not used
A.B.S.	100%	120%	6° at unit natural frequency	Not specified	100%	120%	15° at 10 seconds	None unless alternative criteria is not used
D.N.V.	100%	120%	6° at unit natural frequency	Not specified	120%	120%	Not specified	45m/sec
I.A.C.S.	100%	120%	6° at unit natural frequency	Not specified	Not specified	120%	15° at 10 seconds	Not specified

rack in way of the clamping system, I have already dealt with one aspect of this in the paper under the heading 'Fatigue during tow'. Because during tow the same section of rack is always opposite the clamp it will probably be advisable to machine this section. The clamp itself will of course normally be a machined section.

As far as fatigue in the jacked-up configuration is concerned fatigue of the rack must be considered. Since a mobile rig operates at various water depths, and a change of clamping position by only one or two teeth can make a significant difference, this will not always be a problem. However, if the jack-up is permanently on location at the same water depth, then fatigue considerations will probably be dominant. This aspect has already been dealt with in my reply to P. Fisher.

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Lloyd's Register Technical Association

A REVIEW OF FABRICATION DISTORTION TOLERANCES FOR SHIP PLATING IN THE LIGHT OF THE COMPRESSIVE STRENGTH OF PLATES

by

Prof. Marian Kmiecik
(Guest Lecturer)

Paper No. 6. Session 1986–87

Any opinions expressed and statements made in this paper are those of the author.

A version of this paper was originally published by the Ship Research Institute of the Technical University of Szczecin.

Written contributions to the discussion of this paper are invited from members of the Lloyd's Register Technical Association. To ensure inclusion in the discussion paper, the contributions should be received by the Hon. Secretary in London not later than the 30th June, 1987.

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A REVIEW OF FABRICATION DISTORTION TOLERANCES FOR SHIP PLATING IN THE LIGHT OF THE COMPRESSIVE STRENGTH OF PLATES

by

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SUMMARY

The effect of fabrication distortions and stresses on strength of plates in compression has been analysed. Particular attention has been paid to the effect of the geometry and magnitude of post-welding distortions and the magnitude and distribution of post-welding stresses on ship hull plating.

It has been disclosed that only buckling mode components of fabrication distortions have a detrimental effect on the plating strength. The influence of stresses is different and depends on plate slenderness and on the magnitude of fabrication distortion.

1. NOMENCLATURE

a = plate length
b = plate width
b_e = effective width
e = distance of weld axis from main cross-section axis
E = modulus of elasticity
E_t = tangent modulus
F = stress function or cross-section area of stiffened plate
J = moment of inertia of cross-section of plate with stiffener
k = coefficient
P_u = ultimate load
q = lateral load
q_1 = linear welding energy
R = reduction coefficient
t = plate thickness
t_t = weld dimension
t_w = web thickness of stiffener
T_y = temperature at which linear expansion is equal to yield strain
T_m = temperature to which material is heated
u, v = axial deflections of plate
w_o, $w_{o\,(ij)}$ = initial distortion of plate and components of the distortion
w, $w_{(ij)}$ = elastic deflection of plate and components of the deflection
w_{ob} = buckling mode component of initial distortion
$w_{o\,max}$ = maximum value of initial distortion
$\overline{w}_{o\,max}$ = mean value of $w_{o\,max}$
σ = stress
σ_1, σ_2 = principal stresses
σ_a = axial compressive loading stress
σ_{cp} = Euler's critical compressive stress of plate
σ_e = edge compressive stress
σ_{ef} = effective stress
σ_r = residual compressive stress
σ_{up} = average ultimate stress of plate under compression
σ_t = tensile stress
σ_Φ = standard deviation of load carrying capacity
σ_y = yield stress
ν = Poisson's ratio
η = coefficient of tension width of residual stresses
ϵ = axial strain of plate

$$\epsilon = \frac{u}{a}, \quad \epsilon_c = \frac{\sigma_{cp}}{E}, \quad \epsilon_y = \frac{\sigma_y}{E}$$

μ = coefficient characterising the thermo-physical properties of the material; for low-carbon steel $\mu = -3.13 \times 10^{-6}$ [cm²/cal]

$$D = \frac{Et^2}{12(1-\nu^2)}$$

$$\beta = \frac{b}{t}\sqrt{\frac{\sigma_y}{E}}$$

$$\Phi = \frac{\sigma_{up}}{\sigma_y}$$

ABBREVIATIONS

ABS = American Bureau of Shipping
BV = Bureau Veritas
DNV = Det Norske Veritas
GL = Germanischer Lloyd
LR = Lloyd's Register of Shipping
PRS = Polish Register of Shipping
RZSRR = USSR Register of Shipping

2. INTRODUCTION

Many factors may determine the magnitude of the permissible fabrication distortions of the plating of ships. Among others, these are:

(i) Use of the ship (an excessively corrugated deck plating surface hampers the movement of vehicles, water collects in the corrugation, etc.).
(ii) The aesthetic appearance of the whole ship or of its component elements.
(iii) The optimal economic and organisational technological process.
(iv) The effect of the magnitude of the distortions on the ship's resistance.
(v) The effect of the magnitude of the distortions on strength.

It is obvious that each of these factors affects the magnitude of permissible distortions in a different way. Aesthetic, operational and hydrodynamic considerations dictate minimal distortions while the cheapness of the technological process dictates significantly larger ones.

The present study is mainly devoted to problem (v). In view of this the effect of the magnitude and geometry of welding distortions on axially compressed rectangular plates was analysed. Rectangular plates are the basic structural com-

ponents of the ship plating and compression is the most unfavourable loading if the structure is not perfectly flat.

3. A REVIEW OF EXISTING FABRICATION DISTORTION TOLERANCES FOR THE PLATING OF SHIPS

The unfavourable effect of initial distortions on the strength of axially compressed structural members like beams and columns is a familiar one. The same holds in relation to plates. This factor and those mentioned above are the reason why in Poland and elsewhere shipbuilding, having switched from riveting to welding, limits for the fabrication distortions of ship structures have been introduced. This becomes necessary due to the fact that welding generates very large distortions because of its accompanying thermal processes. When limits are exceeded, straightening is required. Straightening which is performed mainly by heating is a highly laborious operation contributing considerably to an increase in production costs of ships.[11] But regardless of economic losses straightening may exert a harmful influence on the strength properties of the material used in construction of ships and in particular may reduce the resistance of joints to brittle fracture,[7, 36] because it involves changes in the crystal structure of steel and introduces to the structure additional complex stress states as the result of high and often uncontrolled temperatures. In the current rules of the Classification Societies (ABS, BV, DNV, GL, LR, PRS, RZSSR) quantitative tolerances for the maximum permissible distortions are not given. However, when ships are being built the surveyors of these societies impose specific requirements with respect to hull structures on the basis of officially unpublished internal instructions. Shipyards have similar instructions for their own technical control and the examples in Table 1 of maximum permissible distortions within a single spacing illustrate the approach to this problem in Poland and in other countries.[3, 31] In the instructions of the Gdańsk Shipyard permissible deviations are considered in reference to hull regions with a clear distinction between the visible and invisible (there are significantly less strict requirements in relation to structures with a timbering or insulation covering) which follows from the need to ensure a given aesthetic level. The magnitude of the deviations is not dependent either on the thickness of the plating or on other geometrical characteristics of the structure.

Another approach is found in the Gdynia Shipyard in whose instructions permissible distortions decrease with plate thickness for the hull components of main importance for ship strength (shell plating, double bottom and tank top plating, bulkhead plating and strength deck plating). In other items increase of $w_{o\,max}$ is accepted with the increase of t. In the Szczecin Shipyard instruction, deviations are treated in a similar way as in the Gdańsk Shipyard. There is also a distinction between visible and invisible structural elements. However, here permissible deviations increase with an increase in plating thickness.

The tolerances of foreign shipyards are formulated in a way similar to the current requirements in the Gdańsk Shipyard. There is a distinction there between different hull regions and between visible and invisible elements. And apart from American limits where permissible distortions decrease with increase of plate thickness there is no dependence of deviation magnitude on the thickness or other geometrical characteristics of the structure. A characteristic feature of all the tolerances presented is the fact that only the maximum deflection is taken as a criterion as to whether plates qualify for straightening and for post-straightening control, whereas the geometry of the distortion is not investigated or taken into account at all. This procedure is correct from the point of view of the structure's appearance but in the light of the latest research it is not justified in relation to plate strength.

4. POST-WELDING DISTORTIONS AND STRESSES

The phenomenon of shrinkage which accompanies the welding process is the reason why structures which have been fabricated by means of welding are always to a greater or lesser extent deformed and contain balanced internal stresses, which because of their source are also known as post-welding stresses. Shrinkage is a main source of internal stresses which in turn cause deformations of the structures. The magnitude of the deformations and the geometrical shape are a function of the magnitude and distribution of residual stresses. Just like distortions, stresses affect the strength of axially compressed plates. Since they occur simultaneously, an analysis of distortions requires tnat stresses also be taken into account. At present, welding is the dominant process in the joining of steel elements and thus is the main source of fabrication distortion of the stiffened plates of the ship shell-plating and the residual stresses occurring there. For this reason the study will deal first and foremost with welding stresses and deformations treated as synonyms of fabrication stresses and distortions.

Post-welding stresses and distortions are highly varied. Structures with the same geometrical characteristics exhibit significant differences in the magnitude and geometry of their distortions and in the magnitude and distributions of stresses. These differences are the result of deviations in the welding parameters and depend on many uncontrolled factors in the production process as well as on the conditions accompanying the welding process, such as the initial distortion of metal plates and stiffening sections, the boundary conditions and the state of loading of the structure during welding, etc. As a result of this, welding stresses and distortions are treated as random variables whose magnitude can only be estimated on the basis of the appropriate statistical data.

4.1 Post-welding distortions

On the basis of measurements of distortions in the geometrical centre of the surface area of plates framed with stiffeners Faulkner[14] has given the following formulae for the mean value of the maximum deflection:

$$\frac{\overline{w}_{o\,max}}{t} = \begin{cases} k\beta^2 \left(\frac{t_w}{t}\right), & t_w < t \\ k\beta^2, & t_w > t \end{cases} \tag{1}$$

He stated that $k = 0.12$ when $\beta \leqslant 3$ and $k = 0.15$ when $\beta > 3$. In the case of the ordinary ship steel $\sigma_y = 235$ MPa, $\beta = 3$ is equivalent to the slenderness ratio of the plates $b/t = 100$. Formula (1) was obtained on the basis of the regression analysis of about 300 measurements of the bottom plate panels of warships (frigates). Comparing the results obtained with previous results of measurements conducted in Great Britain by the Admiralty Ship Welding Committee[1] Faulkner came to the conclusion that for cargo vessels the coefficient k should also be equal to 0.15 when $\beta \leqslant 3$.

Antoniou[2] has recently come to qualitatively similar conclusions. This study, as the report of Committee III.3 of the 7th International Ship Structures Congress—1979, also contains a detailed discussion of the more important research results on the subject by other authors. Antoniou measured the maximum post-welding distortions of about 1900 plates and found that:

$$\frac{\overline{w}_{o\,max}}{t} = \begin{cases} 0.091\beta^2 \left(\frac{t_w}{t}\right), & t_w < t \\ 0.0628\beta^2 \left(\frac{t_w}{t}\right), & t_w > t \end{cases} \tag{2}$$

for $\beta \leqslant 2.6$, which in the case of the ordinary ship steel corresponds to $b/t \leqslant 90$.

Table 1 Limits of $w_{o\,max}$ (mm) of plating within one spacing in different shipyards and countries

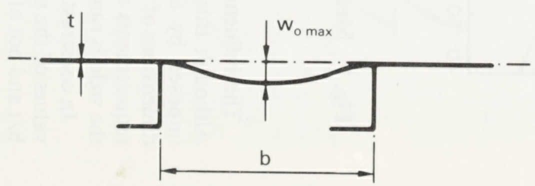
Sub-section	Item	Japan JSQS[1] Standard Range	Japan JSQS[1] Tolerance Limits	Sweden[2]	Federal Republic of Germany[3]	USA[4]	France[5]	Poland Gdańsk Shipyard	Poland Gdynia Shipyard	Poland Szczecin Shipyard
Shell plate	Parallel part side	4	6	6	7–9 max	6,35–3,17	15	6	12–10	5–8
	Parallel part bottom	4	6		7–11 max	6,35–3,17	15	8	12–10	6–9
	Fore and aft part	5	7		7–9 max	6.35–3,17				
Double bottom tank top plate		4	6	6	8–12 max		10–15	8	12–10	7–9
Bulkhead		6	8	6	8–10 max	6,35	15	8	3–7	5–7
Strength deck	Parallel part Between 0,6L	4	6		6–10 max	6,35–3,17	10	7	12–10	4–8
	Fore and aft part	6	9	6	6–10 max		10	7	12–10	4–8
	Covered part	7	9		9–10 max		7–14	12		6–12
Second deck	Bare part	6	8		9	6,35–3,17	10	7	3–7	4–8
	Covered part	7	9		9	9,50–6,35	7–14	12	5–12	6–12
Forecastle deck	Bare part	4	6	6	6	6,35–3,17	8	7	3–7	4–8
Poop deck	Covered part	7	9	10	9	9,50–6,35	7–14	12	5–12	6–12
Superstructure deck	Bare part	4	6		6	6,35–3,17	8	7	12	4–8
	Covered part	7	9	10	9	9,50–6,35	7–14	12		6–12
Cross deck		5	7					7	3–7	4–8
House wall	Outside wall	4	6	6	6	3,17	10	4	12	3
	Inside wall	4	6		6	6,15	10	5		3
	Covered part	7	9	10	8	9,50–6,35	15	10	10	8
Interior member	Web of girder, trans	5	7	6		3,17	10		3–7	
Floor and girder of double bottom		6	8	6		6,35–3,17	10–15		3–7	

NOTE

Increasing numbers mean increase of $w_{o\,max}$ with plate thickness and vice versa.

1. "Japanese Shipbuilding Quality Standard"—1975.
2. "Varvstandard"—1976.
3. "Fertigungsstandard des Deutschen Schiffbaus"—1977.
4. "U.S. Navy Navship. 0900–000–1000 Fabrication, Welding and Inspection of Ship Hull"—1969.
5. "Standards de qualite coque metallique"—1978.

Permissible deflection



It has been stressed in the study, and this is reflected in the formula, that the decisive factor with regard to the magnitude of the distortion is the slenderness ratio b/t of the plate whereas the ratio of the length of the plate to the width of the plate a/b, the thickness of the stiffener web to plate thickness t_w/t and weld thickness to plate thickness t_t/t are less important. It was also noted that a further verification of the influence of the aspect ratio, a/b, on the results obtained would be desirable since the population of plates with (a/b < 2) in the total number of plates investigated was relatively small (190 out of a total of 1908 plates), and that one might expect that in square or nearly square plates (a/b < 2) the magnitude of the distortion would depend not only on longitudinal welds but also on transverse welds. However, it is to be expected that with the currently dominant longitudinal system of stiffening the statistical picture of post-welding distortions in ship's shell plating will be governed mainly by rectangular plates of (a/b > 2).

Measurements carried out in Poland[11, 12] have shown that

$$\frac{\overline{w}_{o\,max}}{t} = 0.00647\left(\frac{b}{t}\right) + 0.0218 \qquad (3)$$

Altogether, 236 plates forming the plating of stiffened panels of the sides, bottoms and decks of cargo vessels of a capacity of 6,300–28,000 dwt were measured.

Geometrical characteristics of the plates investigated:

thickness	t = 7–16 mm
width	b = 500–1000 mm
slenderness	b/t = 42–126
aspect ratio	a/b = 1.0–3.33

As will be seen in Fig. 1, the distortion of plates of ships produced in Poland is decidedly smaller than the values obtained by the above-mentioned investigators, especially in the case of slender plates (b/t > 55) which are especially susceptible to distortion. It seems that this may be attributed to the high quality of hull work in Polish shipyards.

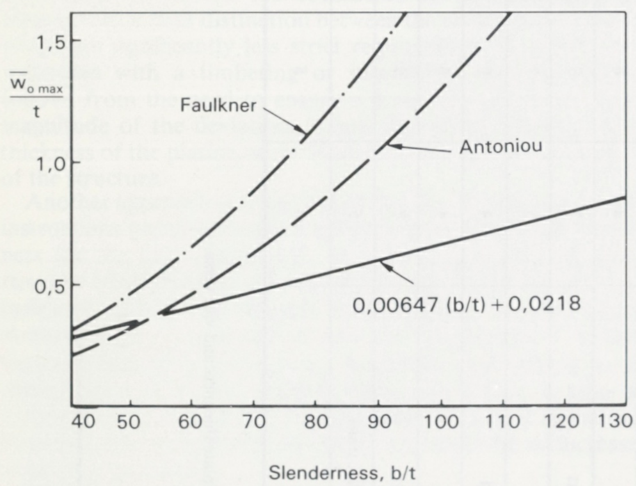


Fig. 1 Mean values of measured maximum post-welding distortion of plates

The influence of t_w/t on distortions was not investigated. Although this parameter is a measure of the requirements imposed by the Classification Societies with regard to the dimensions of welds joining stiffeners to plates, in practice those requirements are exceeded as a rule and the real dimension of the weld is random having no direct relation to t_w/t.

In constrast to formulae (1) and (2), in formula (3) the mean values of the maximum distortions are given in the function of b/t and not of β.

The magnitude of welding distortions, being a result of the transverse and longitudinal shrinkage of the welds and the material depend to a very small extent on the yield point of the material.[33] The elimination of β also makes the application of the formula easier in practice.

In addition to the maximum deviations from flatness, the geometry of the welding distortions was also investigated and approximated by means of a double trigonometrical series of the form:

$$\frac{w_o}{t} = \sum_{i=1}^{m}\sum_{j=1}^{n}\frac{w_{o\,(ij)}}{t}\sin\frac{i\pi x}{a}\sin\frac{j\pi y}{b} \qquad (4)$$

The coefficients of the series were defined on the basis of the recorded deviations from the base plane of twenty points on each of five measurement paths of a plate. The measurement paths are shown in Fig. 2. The figure also contains examples of the shapes of the recorded deformations of three plates of varying aspect ratios and values of the coefficients of the series (4) corresponding to these shapes. This is the first time that investigations of this type (determination of the geometry of welding distortions and not only its maximum deflection) have been carried out in Poland. They do not yet seem to have a foreign counterpart. They were undertaken because earlier theoretical and experimental studies[21, 22, 23, 24, 25, 26, 27] subsequently elaborated in[20, 28, 29, 30, 31, 37, 40] showed that initial distortions only have a negative effect on plate strength when they contain so-called buckling mode components; that is when they are similar to the deflections which occur as a result of the buckling of the perfectly flat plate. When this condition is not met the existence of initial deflections does not have a negative effect on strength.

Apart from the 236 unfaired plates mentioned above, the investigations also included 15 plates subjected to fairing[12]. For faired and unfaired plates with a/b = 2.5–3.5 the share of the detrimental buckling components w_{ob}/t in the total deflection $w_{o\,max}/t$ and the magnitude w_{ob} in relation to plate thickness t was analysed. The results of the analysis are given in Table 2. An increase of the buckling mode component in the distortion remaining after fairing can be seen there. This testifies that the costly and laborious fairing does not generally improve the resistance of plates to compression.

Table 2 Buckling mode component in unfaired and faired plates

	$\overline{w}_{ob}/\overline{w}_{o\,max}$	Standard deviation	$\overline{w}_{ob}/t$	Standard deviation
Unfaired plates	0.180	0.18	0.237	0.28
Faired plates	0.238	0.14	0.257	0.25

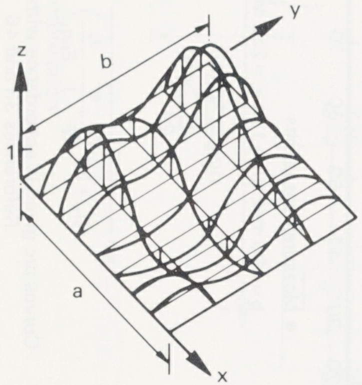
4.2 Post-welding stresses

The determination of post-welding stresses involves much greater difficulties than the determination of distortions. Stress measurements are very difficult, laborious and expensive, especially in natural conditions.[6, 35, 42] This is the reason for the very few measurements which have ever been carried out[17, 34, 39, 44, 45] and measurements carried out on models made in reduced scale do not fully reflect the state of internal strains after welding.

Post-welding stresses and distortions arise simultaneously and are mutually dependent. The magnitude and distribution of

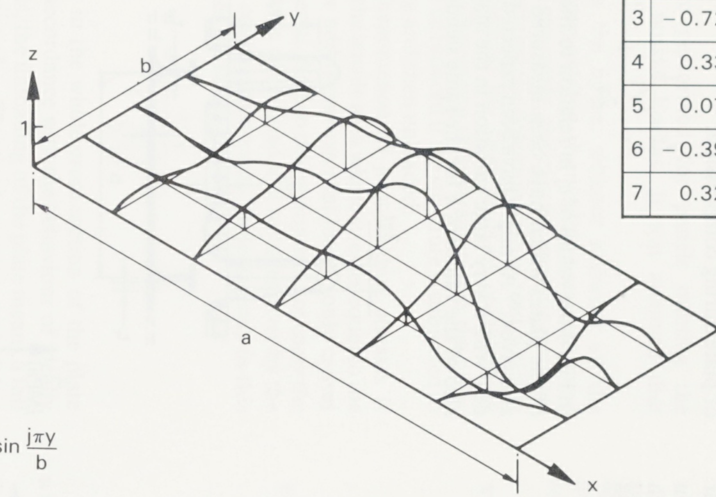
b (mm)	t (mm)	$w_{o\,max}$ (mm)	a/b	b/t	$\frac{w_{o\,max}}{t}$
834	7	3.421	0.96	119.14	0.489

$w_{o(ij)}/t$			
i \ j	1	2	3
1	1.0104	0.9443	0.7758
2	−1.3318	−0.0715	−0.3253
3	0.4473	−0.1134	0.4158
4	−0.4259	0.0129	−0.0270
5	0.4216	−0.0994	−0.1943
6	−0.3556	0.2896	0.0733
7	0.2557	−0.3731	0.0054



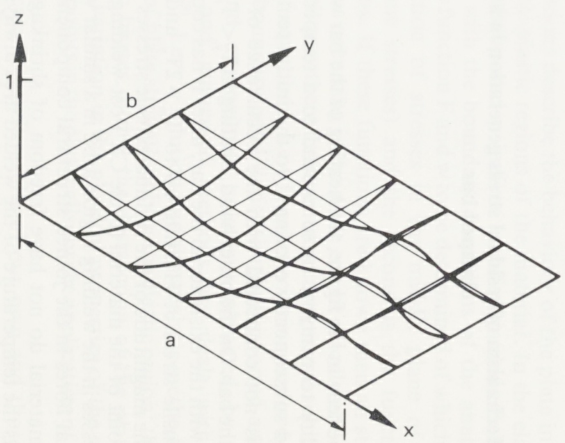
a (mm)	b (mm)	t (mm)	$w_{o\,max}$ (mm)	a/b	b/t	$\frac{w_{o\,max}}{t}$
2,500	700	8	19.840	3.57	87.50	2.480

$w_{o(ij)}/t$			
i \ j	1	2	3
1	0.9972	−0.4873	0.2176
2	0.2999	0.0395	0.0028
3	−0.7242	0.0729	0.1174
4	0.3353	−0.0795	−0.0117
5	0.0756	−0.0066	−0.0593
6	−0.3990	0.0995	0.0799
7	0.3256	−0.0908	−0.0674



$$\frac{w_o}{t} = \sum_i \sum_j \frac{w_{o(ij)}}{t} \sin\frac{i\pi x}{a} \sin\frac{j\pi y}{b}$$

b (mm)	t (mm)	$w_{o\,max}$ (mm)	a/b	b/t	$\frac{w_{o\,max}}{t}$
600	8	2.384	2.33	75.00	0.298



$w_{o(ij)}/t$			
i \ j	1	2	3
1	−0.2048	−0.0219	0.0017
2	−0.1371	0.0059	−0.0079
3	−0.0135	0.0178	0.0048
4	0.0190	−0.0127	0.0028
5	−0.0327	−0.0055	−0.0085
6	0.0140	−0.0097	−0.0103
7	0.0019	−0.0081	0.0062

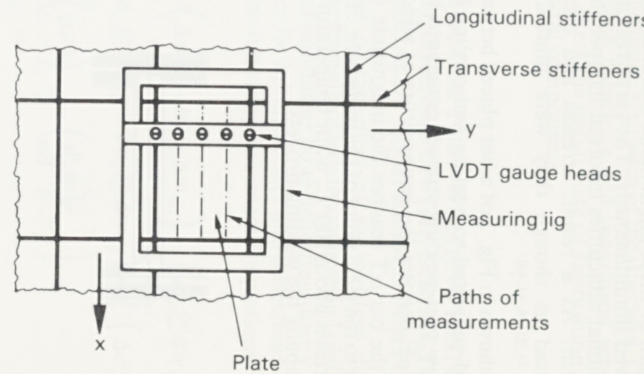


Fig. 2 Examples of measured post-welding distortions of plates

the stresses determines the magnitude and geometry of the distortions. Both of these phenomena and their consequences should be viewed simultaneously. The very great measurement and computational difficulties involved hamper this approach to the problem. As a result, stresses and distortions are usually analysed as though they were mutually independent.(4, 8, 9, 15, 17, 18, 32, 39, 43, 44)

The sketch shown in Fig. 3 of a bar placed between infinitely rigid walls and subjected to heating and cooling helps one to understand how post-welding stresses come into being. In the figure $T_y = \sigma_y/\alpha E$ denotes the temperature at which the linear expansion of the bar αT reaches the magnitude of the strain corresponding to the yield point of the material σ_y. For steel with $\sigma_y = 235$ MPa this is about 100°C, while temperatures exceeding 1000°C accompany the welding process.

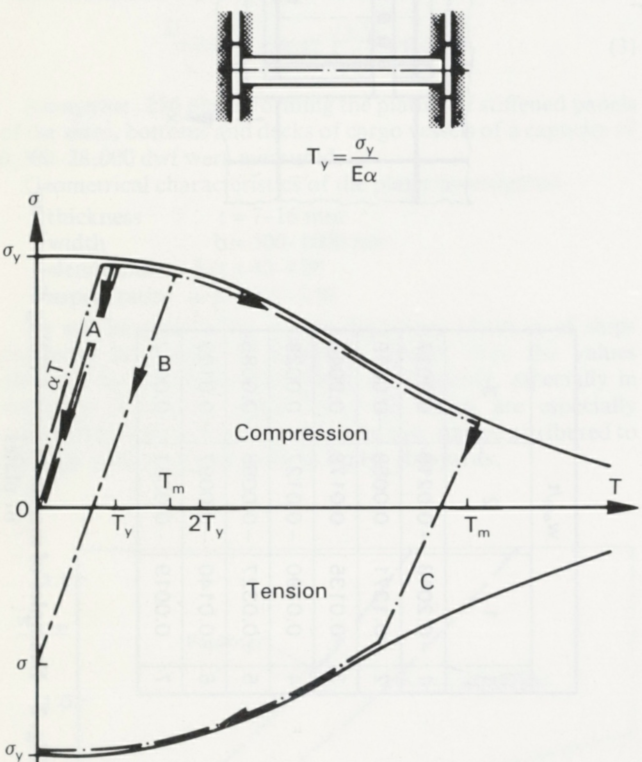


Fig. 3 Mechanism of residual stress generation in a heated clamped bar

As can be seen in the figure, the heating of the bar which has no possibility to contract or to expand to a temperature of $T_m < T_y$ and subsequently cooling it to the initial temperature (cycle A) is not connected with the occurrence of residual stresses in the bar. On the other hand, heating to $T_y < T_m < 2T_y$ is connected with the existence in the bar, after it has cooled, of a residual tensile stress σ. Heating up to $T_m > 2T_y$ and cooling results in the magnitude of the residual tensile stresses reaching the yield point of the material (cycle C). Post-welding residual tensile stresses in the welding region arise in a similar way since the material fibres in the joined structural components and in the weld material do not have freedom of shrinkage during cooling. As the temperature of the welded area always exceeds the temperature T_y, the post-welding stresses in this region are always equal to the yield point of the material.

A typical distribution of post-welding stresses in a plate to which stiffeners have been welded is shown in Fig. 4. To simplify the analysis straight lines are substituted for this distribution in theoretical considerations.

Post-welding stresses as internal stresses are balanced, and since in the tensile region their magnitude is equal to the yield stress of the material (Fig. 4):

$$2\sigma_y \eta t = (b - 2\eta t)\sigma_r$$

Thus the compressive stresses which have an effect on strength of plate during compression:

$$\frac{\sigma_r}{\sigma_y} = \frac{1}{\frac{b/t}{2\eta} - 1} \qquad (5)$$

In Fig. 4 the values of σ_r in relation to the slenderness ratio b/t of the plates which have been measured by various authors is also given.(8) As might have been expected, in practice one has to deal with a very large variation in the magnitude of σ_r and thus also of the values of η for the same slenderness b/t.

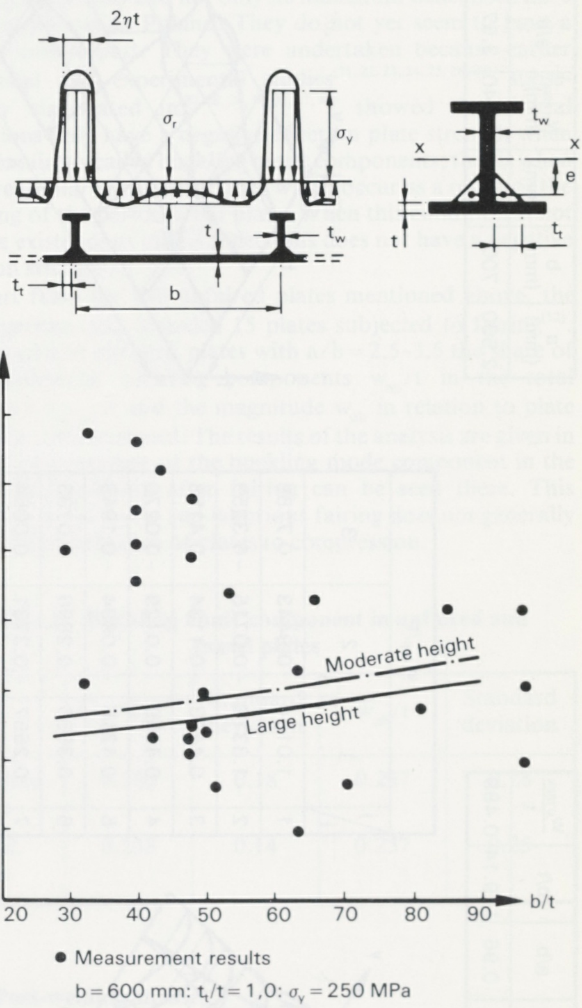


Fig. 4 Post-welding stresses in longitudinally stiffened panels

This is a result of the random character of many of the factors accompanying welding, and especially of the welding parameters, the actual size of the welds and the loading conditions of the structure during welding.

The results of theoretical computations are also given in the figure. In accordance with the theory of deformations of welded structures,[17, 33, 46, 48] for the panel shown in Fig. 4:

$$\eta = \frac{1}{t(t_w + 2t)\left(\frac{1}{F} + \frac{e^2}{J} - \frac{\sigma_y}{E\mu q_i}\right)} \tag{6}$$

Computations were performed for panels meeting the requirements of Det norske Veritas. It was assumed that $q_i = 7250\, t^2_t$ (cal/cm).[33]

As can be seen in the figure, without knowing all the conditions accompanying welding, the welding parameters and the real dimensions of the welds, the existing analytic methods do not allow a sufficiently precise determination of post-welding stresses. Stresses σ_r increase the total loads of compression structures, in this way decreasing their strength. On the other hand these stresses can be reduced if the structure is subjected to external tensile loading and then unloaded.

The shake-down phenomenon of post-welding stresses is explained in Fig. 5. In the tensile area where stresses equal to the yield point prevail, the additional external load σ_t only causes the yielding of the material. And so this region will not carry the load σ_t and thus the whole of the load will be taken over by the $(b - 2\eta t)$ region where compression exists. Subsequently in this area the loading reaches the magnitude of:

$$\sigma'_t = \frac{\sigma_t}{\left(1 - \frac{2\eta}{b/t}\right)}$$

with the mean strains in the whole cross-section of the plate $\epsilon' = \sigma'_t/E$, Fig. 5. In accordance with the behaviour of ideally elastic-plastic materials, the unloading of the plate means a fall in stresses in both regions of σ_t. Thus in the compressed region the residual stresses settle at the level

$$\frac{\sigma'_r}{\sigma_y} = \left(1 - \frac{\sigma_t}{\sigma_y}\right) \frac{1}{\frac{b/t}{2\eta} - 1} \tag{7}$$

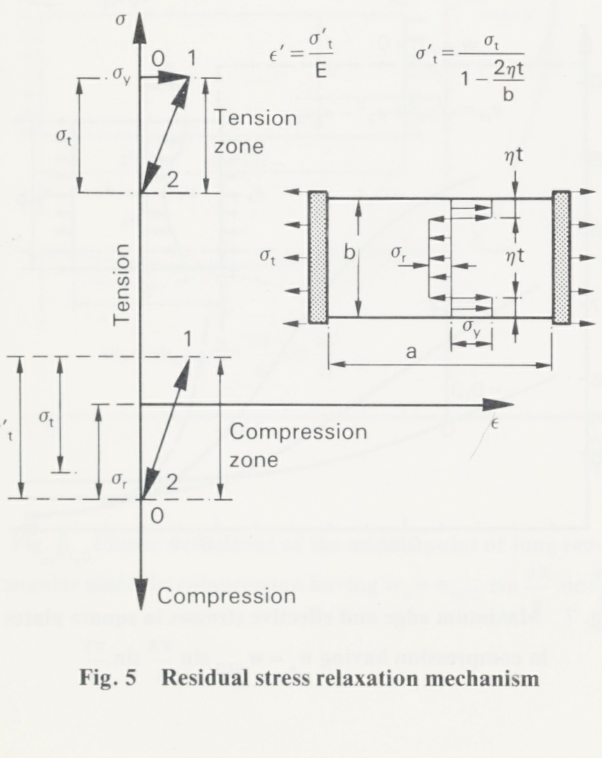


Fig. 5 Residual stress relaxation mechanism

The simplified considerations indicate that the loading of the structure and its subsequent unloading can considerably change the state of the residual stresses of the structure. One might thus expect that both launching and subsequent service of a ship significantly alters the initial state of fabrication stresses in the ship's structural components.

5. THE EFFECT OF POST-WELDING DISTORTIONS AND STRESSES ON THE STRENGTH OF AXIALLY COMPRESSED PLATES

An analysis of the effect of the welding distortions and stresses on the strength of axially compressed plates calls for the application of non-linear large deflection theory of plates with due allowance for initial stresses and deflections. The theory is reflected in the following incremental equations:[29]

$$\begin{aligned}\nabla^4(\Delta F) = E\Big\{ & \left(\Delta w,_{xy}^2 - \Delta w,_{xx}\Delta w,_{yy}\right) - \Big(w_o,_{xx}^{(N)}\Delta w,_{yy} \\ & + w_o,_{yy}^{(N)}\Delta w,_{xx} - 2w_o,_{xy}^{(N)}\Delta w,_{xy}\Big) - \Big[\left(\Delta\epsilon_x^{bp}\right),_{yy} \\ & + \left(\Delta\epsilon_y^{bp}\right),_{xx} - \left(\Delta\gamma_{xy}^{bp}\right),_{xy}\Big]\Big\}\end{aligned} \tag{8}$$

$$\begin{aligned}\nabla^4(\Delta w) = \frac{1}{D}\Big\{ & \Delta q + N_x^{(N)}\Delta w,_{xx} + N_y^{(N)}\Delta w_{yy} + 2N_{xy}^{(N)}\Delta w,_{xy} \\ & + t\Big[\Delta F,_{xx}\left(w_o^{(N)} + \Delta w\right),_{yy} + \Delta F,_{yy}\left(w_o^{(N)} + \Delta w\right),_{xx} \\ & - 2\Delta F,_{xy}\left(w_o^{(N)} + \Delta w\right),_{xy}\Big] - \frac{Et^2}{1-\nu^2}\Big[\left(\Delta m_x^p + \nu\Delta m_y^p\right),_{xx} \\ & + \left(\nu\Delta m_x^p + \Delta m_y^p\right),_{yy} + (1-\nu)\left(\Delta m_{xy}^p\right),_{xy}\Big]\Big\}\end{aligned}$$

where

$\nabla^4(\ \ ) = (\ \ ),_{xxxx} + 2(\ \ ),_{xxyy} + (\ \ ),_{yyyy}$

Δ = increment

$\epsilon_x^{bp}, \epsilon_y^{bp}, \gamma_{xy}^{bp}$ = plastic normal and shear strains, respectively due to membrane effects

m_x^p, m_y^p, m_{xy}^p = plastic normal and shear strains, respectively due to bending effects

$N_x^{(N)}, N_y^{(N)}, N_{xy}^{(N)}, w_o^{(N)}$ = in plane normal and shearing forces and initial deflection, respectively, before $(N+1)$ load increment.

The equations describe the behaviour of the plate in the elastic and elastic-plastic regions of the material. In the elastic range terms with bp and p indices are disregarded. Equations (8) together with the boundary conditions of the analysed plate allow the function F and w to be determined of which the first is the function of stresses in the middle plane of the plate (membrane stresses) and the second is the function of its deflections. If these functions are known then all stresses and deflections occurring in the plate can be calculated.

Up until now there has been no exact general solution of this system of equations. Levy's exact solution for two particular cases of boundary conditions (simply supported and clamped edges) and approximate solutions obtained by means of analytic and numerical methods are dealt with in more detail in[29]. Levy's solution and the analytic methods allow an analysis of plate strength only in the elastic region; the creation of plastic hinges during the destruction of the plate and the resultant variation in the physical properties of the material in different points in the plate calls for the application of numerical methods.

Significant research results of our own and those of other authors obtained by analytical and numerical methods are presented hereafter.

5.1 The effect of distortions on strength in the elastic range[20, 21, 23, 24, 26, 31, 37]

In Fig. 6 the elastic deflections of an initially distorted plate (solid line) is shown against deflections of a perfectly flat plate (broken line). The dimensions of the plate are $a/b = 1$ (square plate); initial deformation is in the form of regular half-wave:

$$w_o = w_{o(11)} \sin \frac{\pi x}{a} \sin \frac{\pi y}{b} \tag{9}$$

As will be seen in the figure, if a plate does not have initial distortion it remains flat $w/t = 0$ until the axial loading σ_a reaches its critical value σ_{cp}. When this value is exceeded ($\sigma_a/\sigma_{cp} > 1$) the plate buckles ($w/t \neq 0$), although further load increase is not accompanied by an unlimited increase of deflection. The reason for this is the supported edges of the plate parallel to the direction of the action of the axial load. These do not buckle and they take over that part of the load which cannot be borne by the central buckled section of the plate. This causes a change in the distribution of the compressive stresses in the plate: stresses at the edges increase and decrease in the centre of the plate (Fig. 7). Thus the plate can still be loaded and at a certain level of axial load in places where the strain of the material is greatest, the first yielding will occur. With a further load increase the yielding spreads, which leads to the collapse of the plate. Thus failure does not occur at the critical load but at the load which causes the formation of significant yield zones in the plate. The difference between the critical buckling load and the ultimate load increases for an increase in the ratio of the material yield point of the plate to its critical stress, that is, the higher the ratio b/t. In aircraft structures where there are very slender plates with large b/t ratios the ultimate loads may be up to thirty times greater than critical loads. In ship structures with a slenderness ratio b/t most frequently within the limits 35 to 100, the ratio of the ultimate load σ_{up} to the critical load σ_{cp} is within the range 1 to 1.8.

The initial distortion changes the behaviour of the plate but —and this should be emphasised—only in the range of axial loads close to and less than critical loads. With loads significantly greater than buckling loads ($\sigma_a/\sigma_{cp} > 1.2$) the deflections of an initially distorted plate approximate those of the ideally flat plate. Thus the level of stresses in these two cases do not differ much after the critical state has been exceeded. This can be seen in Figs 6 and 7 where the maximum deflections and the maximum edge and effective stresses are given for various values of σ_a/σ_{cp}.

The case analysed above and repeatedly discussed in the available literature on the subject[4, 13, 14, 15, 17, 43, 47] and which constitutes the classical approach to the evaluation of the effect of initial distortions on the strength of axially compressed plates does not comprehensively exhaust this problem. This is because:

(i) rectangular plates exist apart from square plates and occur more frequently in practice, e.g. in the longitudinally stiffened plating of modern ships the ratio a/b is usually equal to, or greater than, three,

(ii) the welding distortions of plates are characterised by a high degree of irregularity and the shape of these distortions cannot be approximated by a single half-wave since it is usually the sum of several terms of the series (4), as can be seen in Fig. 2.

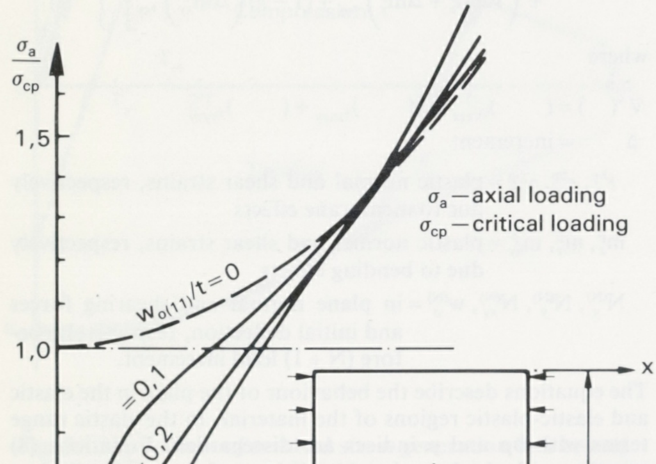


Fig. 6 Maximum elastic deflections of square plates in compression having initial deformation $w_o = w_{o(11)} \sin \frac{\pi x}{a} \sin \frac{\pi y}{b}$

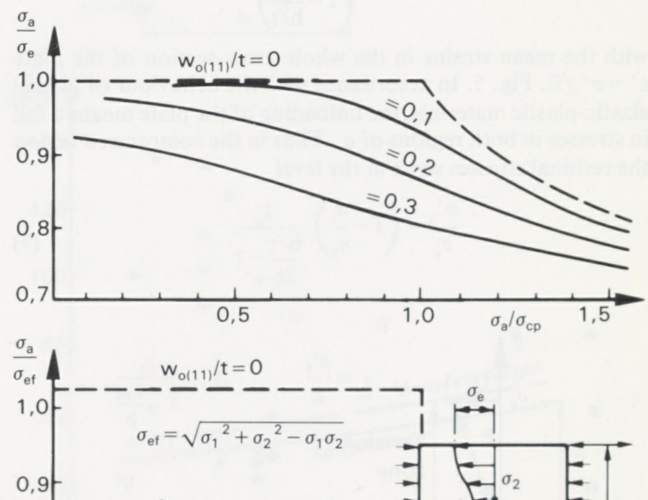


Fig. 7 Maximum edge and effective stresses in square plates in compression having $w_o = w_{o(11)} \sin \frac{\pi x}{a} \sin \frac{\pi y}{b}$

The effect of the aspect ratio a/b and the shape of the initial distortions on plate strength is illustrated in Figs 8, 9 and 10. It follows from the figures that there is no difference in the magnitude of deflections and stresses in only two ideally flat plates if one of them has $a/b = 1$ and the other $a/b = 3$. This results from the fact that a simply supported rectangular plate whose length is three times greater than its width loses stability in the form of three regular half-waves, thus resolving into three square plates, the state of stresses in each of the plates created in this way being exactly the same as in the single plate where $a/b = 1$. Significant differences in the state of stresses will occur when both plates are initially distorted. The same kind of initial deformation in the form of regular half-waves which reduced the strength of the square plate clearly increases the strength of the rectangular plate. For in accordance with Figs 9 and 10, in the range of over-critical loads plates with $a/b = 3$ have smaller stresses than when they have no deformations. The situation only deteriorates when one of the components of the initial distortion corresponds to the deflections of the ideally flat plate after loss of stability. As can be seen in Figs 11, 12 and 13, the stresses in plates with a dimension of $a/b = 3$ assumes values greater than in ideally flat plates on condition that there exists a buckling mode component $w_{o(31)}$ and that it is sufficiently large. This means that not only the magnitude but also the geometry of the deformation affects the strength of the plate. Here, as far as the deterioration of the state of stresses is concerned the second factor is decisive.

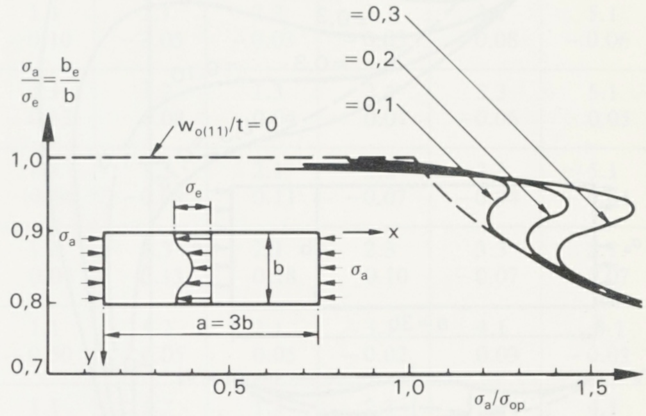


Fig. 9 Effective width of long rectangular plates in compression having $w_o = w_{o(11)} \sin \frac{\pi x}{a} \sin \frac{\pi y}{b}$

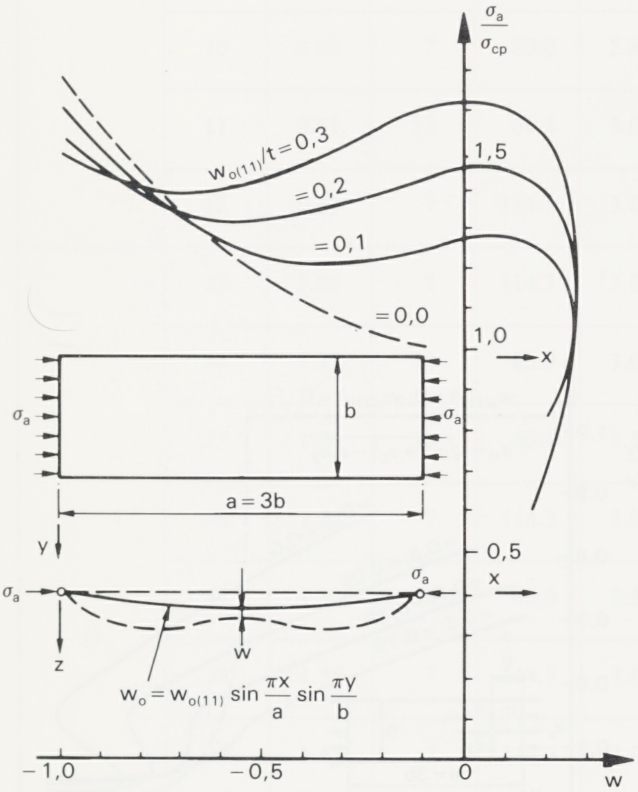


Fig. 8 Elastic deflections of the middle point of long rectangular plates in compression having $w_o = w_{o(11)} \sin \frac{\pi x}{a} \sin \frac{\pi y}{b}$

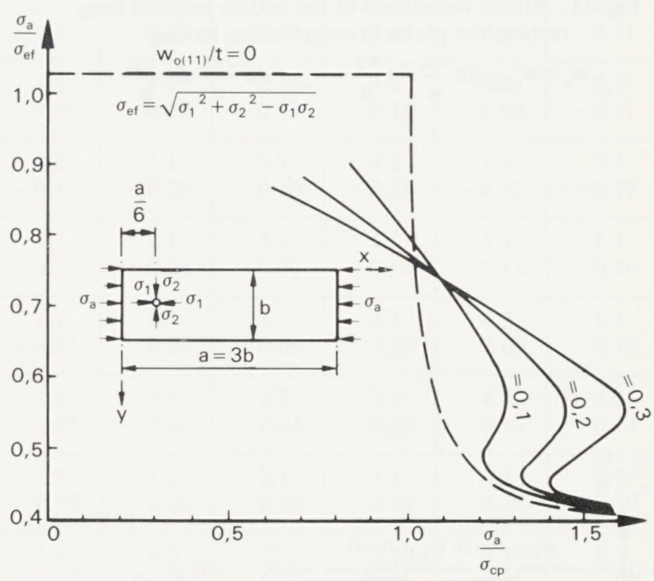


Fig. 10 Maximum effective stresses in long rectangular plates in compression having $w_o = w_{o(11)} \sin \frac{\pi x}{a} \sin \frac{\pi y}{b}$

As has already been mentioned, in shipbuilding practice the only criterion accepted for qualifying plates for straightening and post-straightening control is the maximum fabrication distortion whereas the geometry of the distortion is not checked or taken into account at all. In order to verify whether this approach to the actual plating of a ship is justified, computations of the maximum stresses of 22 plates[31] were carried out on the basis of the results of the measurements of the actual welding distortions of the plates.[11, 12] The dimensions of the plates, the values of the maximum welding distortions and the values of the coefficients of the double trigonometric series (4) which are the reflections of the geometry of the actual welding distortions are given in Table 3. The plates were divided into two groups of a maximum relative distortion of $w_{o\,max}/t<1$ and $w_{o\,max}/t>1$. As follows from Table 4, the behaviour of the axially compressed plate does not, in the whole range of axial loads, depend explicitly on the magnitude of the maximum relative initial distortion $w_{o\,max}/t$. In spite of the large variation in $w_{o\,max}/t$, the computed values of the maximum compressive stresses at the plates longitudinal edges and the maximum effective stresses lie close to each other for both groups of plates. This is illustrated by the arithmetical means of the computed values for these groups. The means do not vary more than 16 per cent with a variation in distortion of 240 per cent.

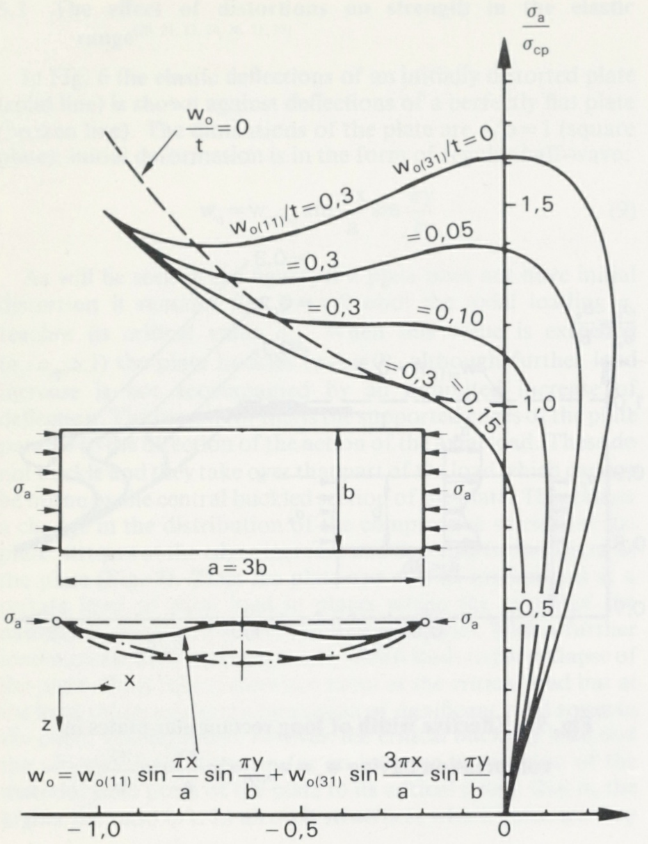


Fig. 11 Elastic deflections of the middle point of long rectangular plates in compression having

$$w_o = w_{o(11)} \sin\frac{\pi x}{a} \sin\frac{\pi y}{b} + w_{o(31)} \sin\frac{3\pi x}{a} \sin\frac{\pi y}{b}$$

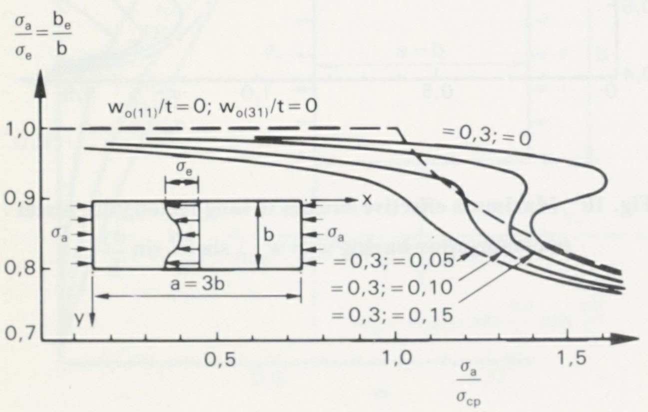


Fig. 12 Effective width of long rectangular plates in compression having

$$w_o = w_{o(11)} \sin\frac{\pi x}{a} \sin\frac{\pi y}{b} + w_{o(31)} \sin\frac{3\pi x}{a} \sin\frac{\pi y}{b}$$

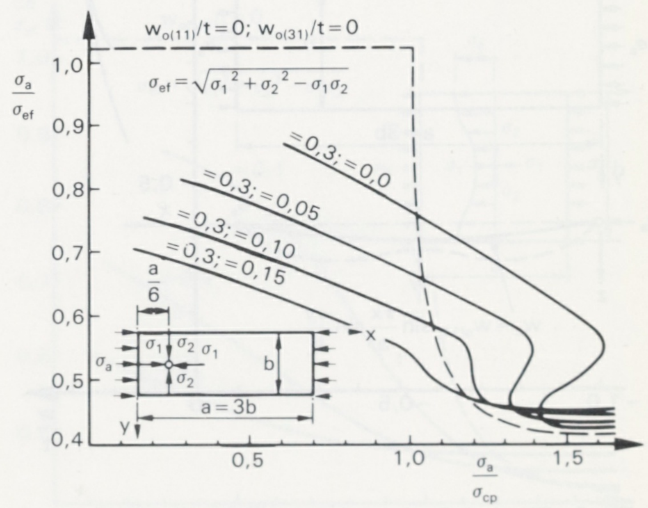


Fig. 13 Maximum effective stresses in long rectangular plates in compression having

$$w_o = w_{o(11)} \sin\frac{\pi x}{a} \sin\frac{\pi y}{b} + w_{o(31)} \sin\frac{3\pi x}{a} \sin\frac{\pi y}{b}$$

Table 3 Data of plates having post-welding distortions

Group	Plate No.	$\frac{w_{o\,max}}{t}$	t (mm)	$\frac{b}{t}$	$\frac{a}{b}$	Indices of series components (ij)					
						Values of $w_{o\,(ij)}/t$					
I	1	0.17	13	61.5	3.0	1.1 −0.06	1.2 −0.05	1.3 −0.02	3.1 −0.09	4.1 −0.12	5.1 −0.02
	2	0.22	13	63.1	3.0	1.1 −0.10	2.1 −0.05	2.2 −0.03	2.3 −0.03	3.1 −0.08	5.1 −0.06
	3	0.31	7	124.4	1.0	1.1 0.13	1.2 0.08	1.3 0.04	2.1 0.07	3.3 −0.06	5.1 0.05
	4	0.34	7	123.3	1.0	1.1 0.04	1.3 −0.09	2.1 0.11	2.2 −0.07	3.3 −0.04	5.1 −0.04
	5	0.45	7	80.0	3.0	1.1 0.04	1.3 −0.13	2.1 0.18	2.3 0.10	3.3 −0.07	5.1 −0.07
	6	0.49	13	61.5	3.0	1.1 −0.50	1.2 0.05	2.1 0.05	3.1 −0.02	4.1 0.03	5.1 −0.03
	7	0.49	7	119.1	1.0	1.1 −0.14	1.2 −0.14	1.3 −0.11	2.1 0.19	3.1 −0.06	4.1 0.06
	8	0.50	10	60.0	3.5	1.1 −0.09	1.2 −0.09	2.1 0.05	3.1 −0.20	4.1 −0.18	5.1 −0.21
	9	0.60	10	60.0	3.5	1.1 −0.45	2.1 −0.03	3.1 −0.14	4.1 0.02	4.2 0.02	5.1 −0.02
	10	0.60	7	80.0	2.89	1.1 0.25	1.2 0.23	1.3 −0.14	2.1 0.21	4.1 0.12	4.2 −0.11
	11	0.61	13	61.5	3.0	1.1 0.34	1.3 −0.08	2.1 −0.15	3.1 0.13	4.1 −0.09	5.1 −0.11
	12	0.63	7	114.3	3.0	1.2 −0.17	2.1 −0.28	3.1 −0.10	4.1 −0.15	5.1 −0.12	7.1 −0.17
II	13	1.05	7	114.3	3.0	1.1 −0.38	2.1 0.24	3.1 −0.60	4.1 −0.55	5.1 −0.11	7.1 0.10
	14	1.15	10	80.0	3.0	1.1 −0.47	2.1 0.27	3.1 0.08	4.1 −0.31	4.3 0.09	5.1 0.13
	15	1.20	10	60.0	3.5	1.1 1.05	2.1 −0.02	2.2 −0.04	3.1 0.05	4.1 −0.12	5.1 0.14
	16	1.24	7	114.3	3.0	1.1 0.52	1.2 0.26	2.1 0.27	3.1 0.39	4.1 −0.25	6.1 0.40
	17	1.30	10	60.0	3.5	1.1 1.03	1.2 −0.48	2.1 −0.17	3.1 −0.11	4.1 0.20	5.1 −0.08
	18	1.35	7	114.3	3.0	2.1 0.80	3.1 0.31	3.3 0.18	4.1 0.19	5.1 0.20	6.1 0.15
	19	1.47	7	114.3	3.0	1.1 −0.50	2.2 −0.20	3.1 0.80	3.2 −0.16	5.1 −0.22	6.1 0.15
	20	1.60	9	96.8	2.0	1.1 −1.48	1.2 0.10	1.3 −0.13	2.1 −0.02	3.1 −0.53	5.1 0.14
	21	1.95	9	57.8	4.0	1.1 1.7	1.2 0.65	2.1 0.30	3.1 −0.06	4.1 −0.02	5.1 −0.02
	22	2.93	9	77.8	3.6	1.1 −1.51	1.2 0.21	2.1 1.09	3.1 −0.22	4.1 0.50	5.1 −0.03

Table 4 Mean values of maximum distortions of maximum edge stresses and of maximum effective stresses of plates given in Table 3

Group	$\frac{\overline{w}_{o\,max}}{t}$	$\frac{\sigma_a}{\sigma_{cp}}$	$\left(\frac{\overline{\sigma}_a}{\sigma_e}\right)_{max}$	$\left(\frac{\overline{\sigma}_a}{\sigma_{ef}}\right)_{max}$
I n = 12	0,451	0,6	0,798	0,620
		1,0	0,814	0,557
		1,5	0,738	0,507
II n = 10	1,524	0,6	0,733	0,640
		1,0	0,705	0,602
		1,5	0,667	0,559

Fig. 14 is equally demonstrative in this respect. As can be seen in the figure with a particular welding distortion geometry the state of stresses in a plate of $w_{o\,max}/t = 1.17$ may be more favourable than in a plate where $w_{o\,max}/t = 0.17$. Straightening which reduces the maximum initial deflection by about 50 per cent does not lead to an improvement but to a significant deterioration of the stress state when the geometry of distortion after straightening is unfavourable.

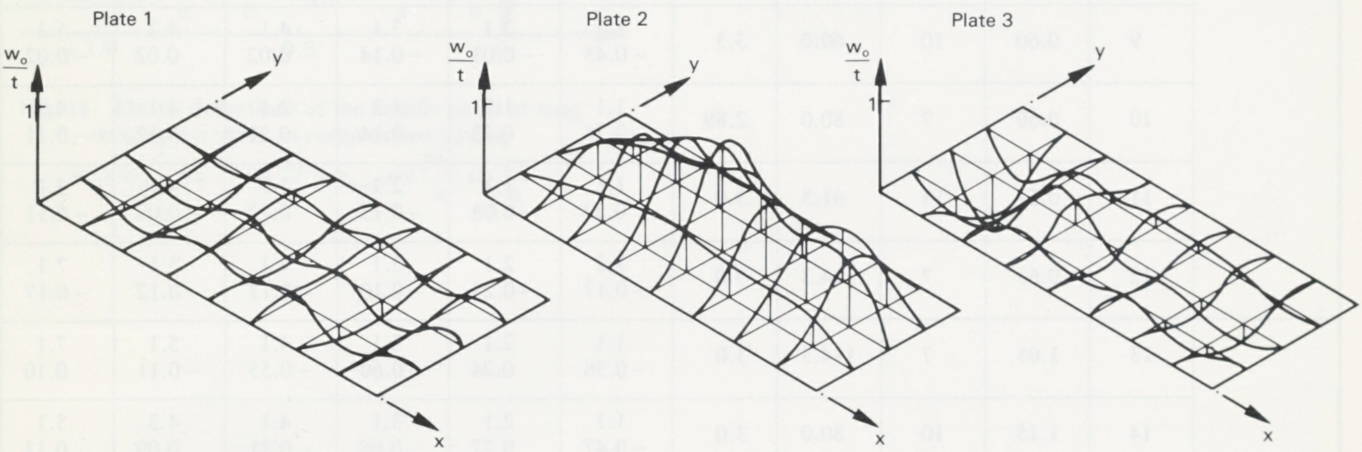
5.2 The effect of distortions on the ultimate strength of plate[4, 5, 8, 9, 10, 14, 15, 16, 17, 18, 28, 29, 30, 32, 40, 41, 43]

The elastic-plastic nature of the collapse of plates requires that numerical methods be applied for the solution of equations (8). These methods alone allow an analysis of plates taking into account the geometrical non-linearity (large deflections) and physical non-linearity (yielding) which accompany the exhaustion of the load-carrying capacity of the plates.

Among the most widely used of the existing numerical methods are the finite difference method and the finite element method. An important advantage of these methods is their versatility. They may be used for any boundary conditions and for all kinds of non-linear phenomena. Their disadvantage is the large amount of programming and computation they require. Thus, an effective solution of the non-linear problems of plates is only possible with the use of large and fast computers.[29]

It seems that computational difficulties in 1975 led Faulkner[14] to propose (on the basis of a statistical analysis of experimental tests) the following semi-empirical formula for the ultimate strength of simply supported plates without residual stresses and with moderate fabrication distortions:

$$\Phi = \frac{\sigma_{up}}{\sigma_y} = \frac{2}{\beta} - \frac{1}{\beta^2} \quad \text{for} \quad \beta \geq 1$$
$$\Phi = 1 \quad \text{for} \quad \beta < 1 \tag{10}$$



σ_a = axial loading
σ_{cp} = critical stress of plate
σ_{ef} = effective stress

$\sigma_{ef} = \sqrt{\sigma_1^2 + \sigma_2^2 - \sigma_1\sigma_2}$

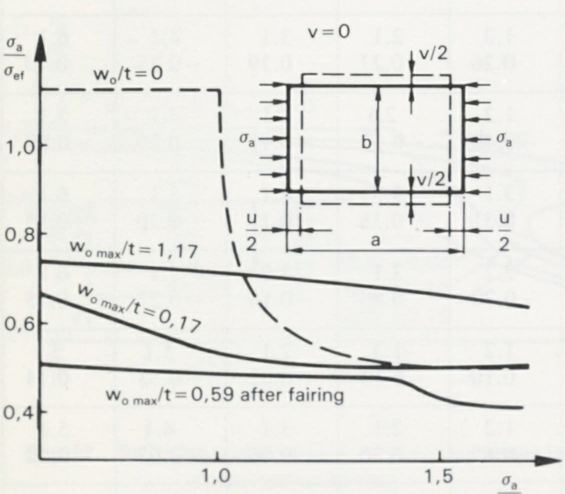


Plate No.	$\frac{a}{b}$	t (mm)	$\frac{b}{t}$	$\frac{W_{o\,max.}}{t}$	Indices of series components (i j) / Values of $W_{o(i\,j)}/t$						Note
1	3	13	61.5	0.17	1.1	1.2	1.3	3.1	4.1	5.1	
					−0.06	−0.05	−0.02	−0.09	−0.03	−0.02	
2	3.5	10	60	1.17	1.1	2.1	1.3	3.1	4.1	5.1	
					0.86	−0.05	−0.19	0.05	−0.10	0.10	
3	3.5	10	60	−0.59	1.1	1.2	3.1	4.1	5.1	6.1	After fairing
					−0.04	−0.09	−0.17	−0.11	−0.17	−0.07	

Fig. 14 Maximum effective stresses in plates in compression having post-welding distortions

Fig. 15 shows the Φ curves against the Euler critical stress and the experimental results. Equation (10) defines the mean value of the experimentally determined ultimate strength with a standard deviation of

$$\begin{aligned} \sigma_\Phi &= 0.05 \quad \text{for} \quad 0.5 \leqslant \beta \leqslant 2.5 \\ \sigma_\Phi &= 0.02\beta \quad \text{for} \quad \beta > 2.5 \end{aligned} \tag{11}$$

Equation (10) was also verified during tests of the load carrying capacity of ship hatch covers.[28, 30] Three full scale hatch covers of the dimensions 12240 × 2935 × 536 mm were tested. There was a satisfactory agreement between the measurement results and the calculations on the basis of equation (10).

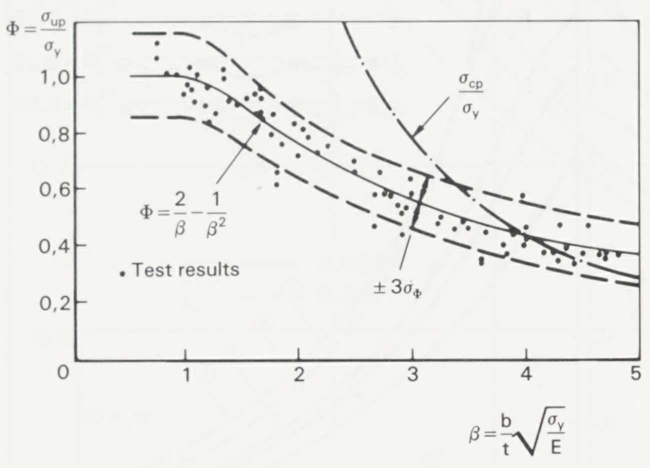


Fig. 15 Ultimate strength of simply supported square plates in compression without residual stresses and with moderate distortions

Fig. 16 gives the results obtained by Ueda and others using the finite element method.[43] They analysed simply supported square plates with initial distortions in the form of a regular half-wave described by equation (9). In accordance with Fig. 16 the results obtained fall in fact in the range determined by three standard deviations within which would lie 99,7 per cent of all experiments analysed by Faulkner,[14]

$$\Phi_{\sigma\Phi} = \Phi \pm 3\sigma_\Phi \tag{12}$$

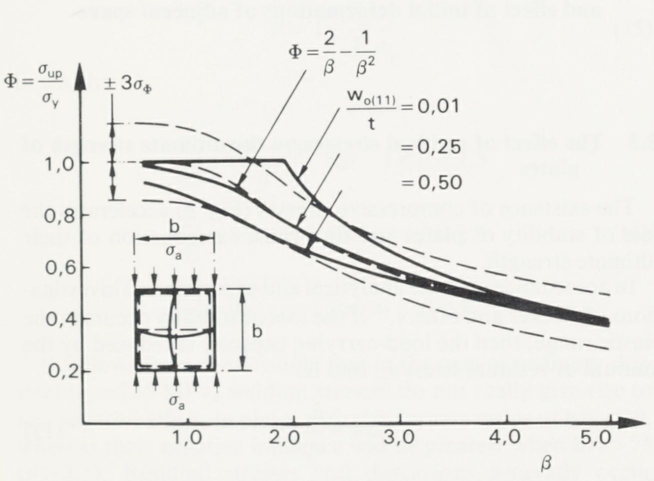


Fig. 16 Ultimate strength of simply supported square plates having $w_o = w_{o(11)} \sin \frac{\pi x}{a} \sin \frac{\pi y}{b}$

The negative effect of initial distortions on the load-carrying capacity can be seen, and the smaller the slenderness ratio, the greater is the influence since the buckling mode of a square plate is the same as the assumed initial deflection.

Reference (43) also contains an analysis of the effect of geometry of initial deflections on the load carrying capacity, but only for square plates. Fig. 17 gives the maximum plate deflections as a function of axial loads with four different shapes of initial deflection and the same values of $w_{o\,max}/t = 0.01$ and 0.5. The variation in the load-carrying capacity can be clearly seen, although for all of the initial deformations it is lower than the load-carrying capacity of the plate with no distortion. However, it should be noted that in each of the initial deformations the buckling mode component of the square plate was dominant.

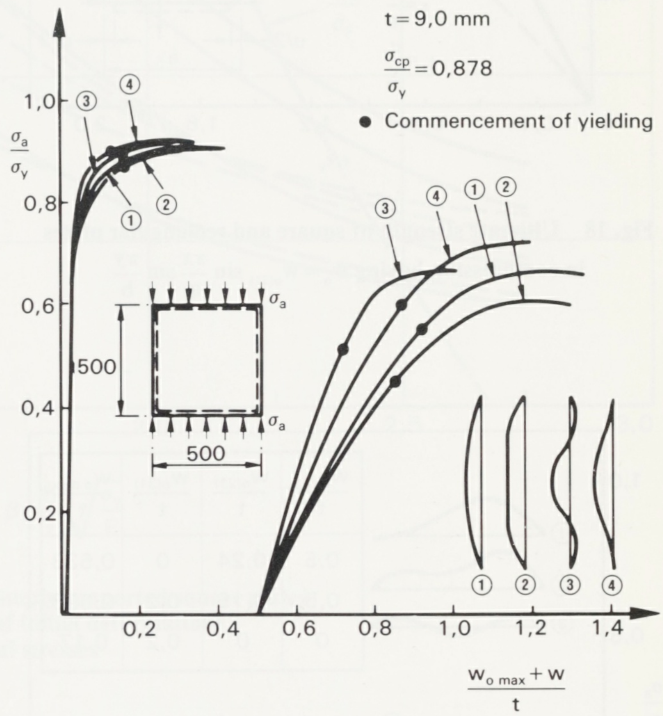


Fig. 17 Maximum deflections of simply supported square plates in compression having distortions of different shapes and with $w_{o\,max} = 0.01t$ and $0.5t$

The load-carrying capacity of rectangular plates with an initial deflection which did not accord with the buckling mode was investigated by Frieze and others[18] and Carlsen and Czujko.[10] (The latter, in collaboration with the present author using the measurements of the actual distortions of the plates given in[11, 12]). In both cases the calculations were based on the finite difference method. Fig. 18 contains the results of the investigations of Frieze and others and Fig. 19 those of Carlsen and Czujko. Since in case of a/b = 3 the buckling of the plate takes the shape of three half-waves, the initial deflection of one half-wave does not lower but raises the load-carrying capacity of the plate, in contrast to the case with the square plate (Fig. 18).

The calculations of Carlsen and Czujko fully confirmed the results of the analysis in the elastic range.[31] Only the buckling component lowers the load-carrying capacity of the plate. Initial distortions which do not contain a buckling component are not detrimental, regardless of their size; they do not reduce but increase the load-carrying capacity of the plate (Fig. 19).

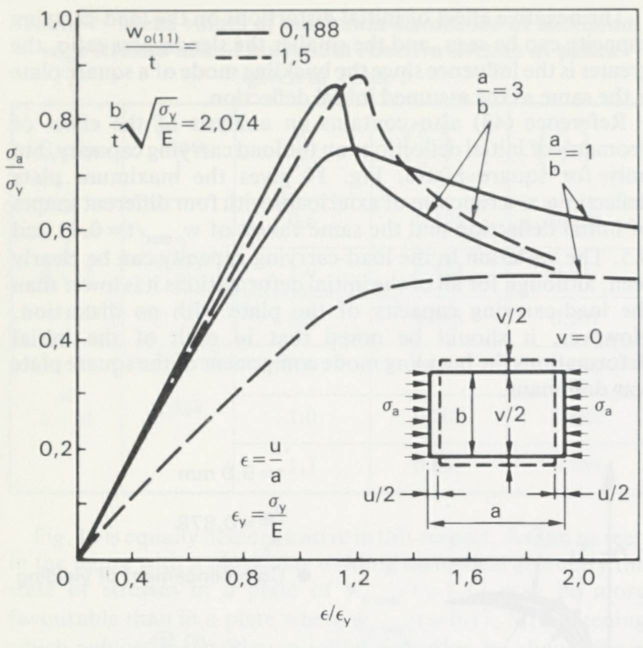


Fig. 18 Ultimate strength of square and rectangular plates in compression having $w_o = w_{o(11)} \sin \frac{\pi x}{a} \sin \frac{\pi y}{b}$

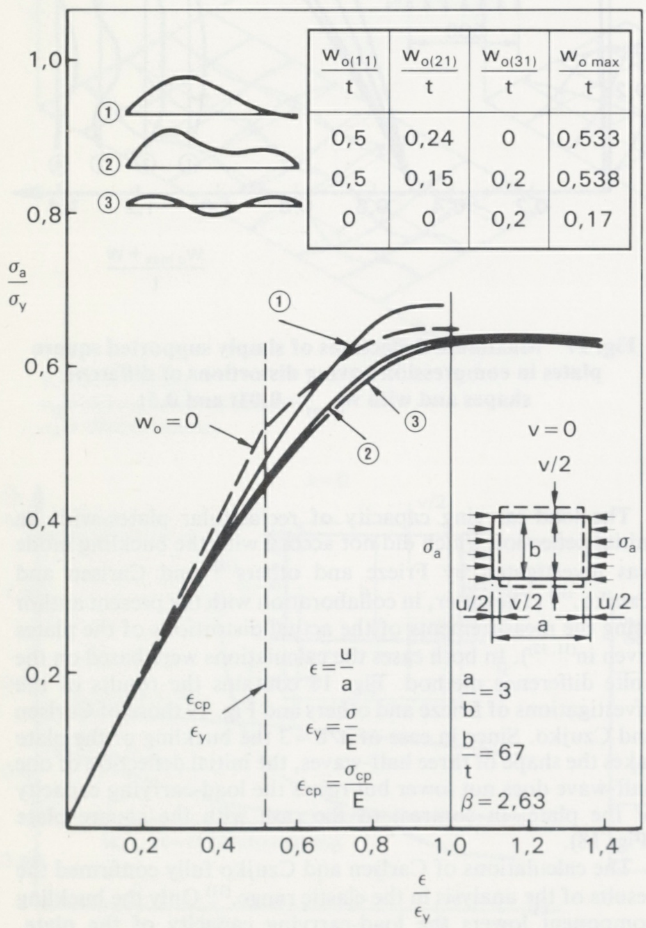


Fig. 19 Effect of shape of initial deformation on ultimate strength of a rectangular plate in compression

The positive effect of initial distortions which are not in accordance with the buckling mode also occurs when several neighbouring plates are simultaneously in compression. The welding distortions of plates which are adjacent and separated by stiffeners are as a rule of the same form, while the sign of their deflections caused by buckling are different. Hence the reinforcing effect of the initial deflections of the plates shown in Fig. 20.[10, 32]

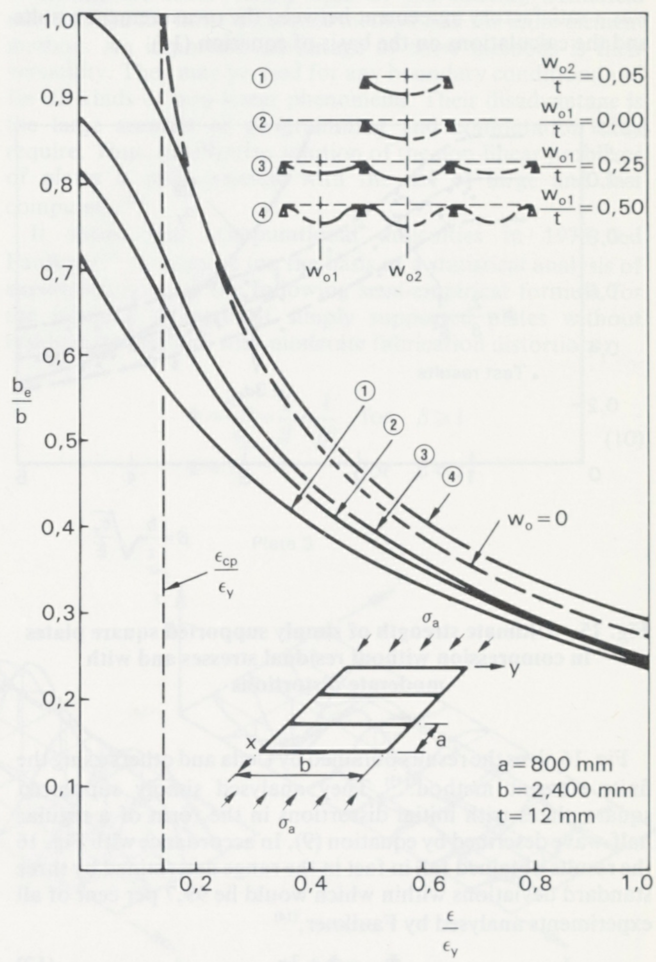


Fig. 20 Effective width of plate in transverse compression and effect of initial deformations of adjacent spans

5.3 The effect of residual stresses on the ultimate strength of plates

The existence of compressive stresses (Fig. 4) accelerates the loss of stability of plates and thus causes a reduction of their ultimate strength.

In accordance with the analytical and experimental investigations of Becker and others,[4] if the loss of stability occurs in the elastic range, then the load-carrying capacity is reduced by the amount of residual stress σ_r, that is:

$$\Delta\Phi = \frac{\sigma_r}{\sigma_y} \tag{13}$$

On the other hand, with buckling in the elastic-plastic range:

$$\Delta\Phi = \frac{E_t}{E} \cdot \frac{\sigma_r}{\sigma_y} \tag{14}$$

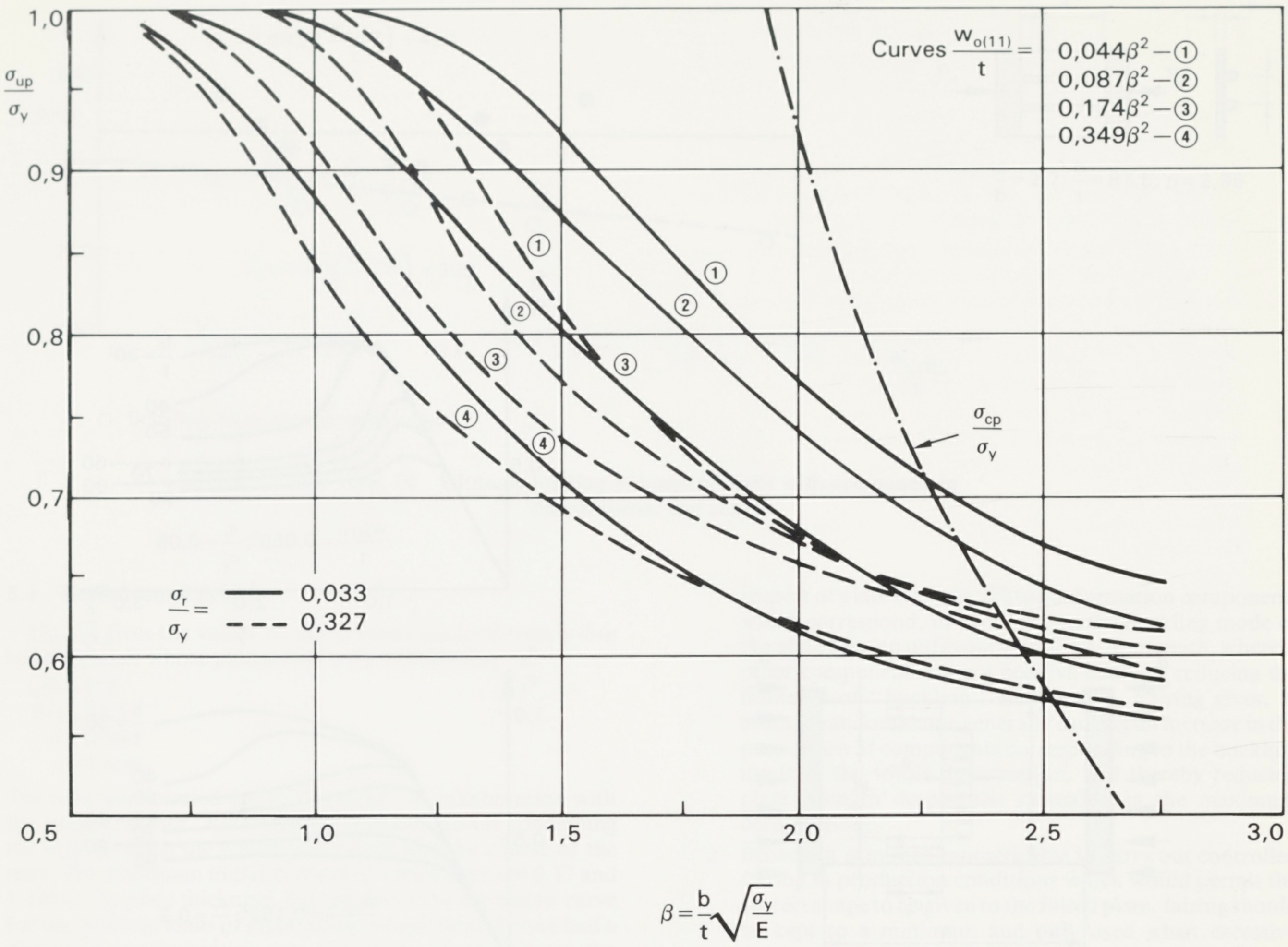


Fig. 21 Ultimate strength of simply supported square plates in compression: effect of initial deformations and residual stresses

In this connection, Faulkner[15] introduced the following reduction coefficient into his formula (9):

$$R=1-\frac{E_t}{E}\frac{\sigma_r}{\sigma_y} \tag{15}$$

in which:

$$\frac{E_t}{E}=\frac{2(\beta-1)}{\beta} \quad \text{for} \quad 1\leqslant\beta\leqslant 2.5$$

$$\frac{E_t}{E}=1 \quad \text{for} \quad \beta>2.5 \tag{16}$$

$$\frac{E_t}{E}=0 \quad \text{for} \quad \beta<1$$

It follows from the formula that in the case of ordinary ship steel ($\sigma_y=235$ MPa) welding stresses do not really give rise to any negative effects in plates with slenderness ratios of $b/t<30$, whereas their negative influence will be greatest when $b/t>75$ ($\beta>2.5$). Residual stresses and distortions generally occur simultaneously and so, in the latest investigations[17, 18, 32, 40, 43] based on numerical methods, they have been dealt with together.

In Fig. 21 the ultimate strength of plates with initial stresses and deformations is given.[18] Practically no effect of the residual stresses on the ultimate strength of plates with sufficiently large initial deformations is observed. This seemingly surprising result can be explained by the analysis of the behaviour of the compressed plate shown in Fig. 22 on the assumption that it will not buckle.

The compression of a plate without residual stresses will not change the elastic-plastic nature of the material and will follow the OAB curve. The existence of residual compressive stresses accelerates the yielding of the material in the region $(b-2\eta t)$ causing a sudden drop in the stiffness of the plate in the ratio of $2\eta t/b$ at a load of $\sigma=\sigma_y-\sigma_r$. As a result, curve OAB changes to OA′B′B. It is characteristic that the load-carrying capacity of the plate does not change, remaining σ_y if AB is flat. The drop in load-carrying capacity occurs when the initial behaviour of the plate is in the form of a curve with a clearly marked extreme point (curve OA″C), since there then occurs a "cutting off" of the extreme point of this curve. The existence of initial distortions "flattens" the curves of compressed plates, hence the minimal influence of compressive residual stresses when simultaneously significantly large distortions exist. This is particularly relevant to plates when $b/t>40$ because when $b/t<40$ the curves σ–ϵ in the region of the maximum load-carrying capacity of the plate are flat and plates of that slenderness ratio are not sensitive to residual stresses, even when there are no deformations Fig. 23.[40, 41]

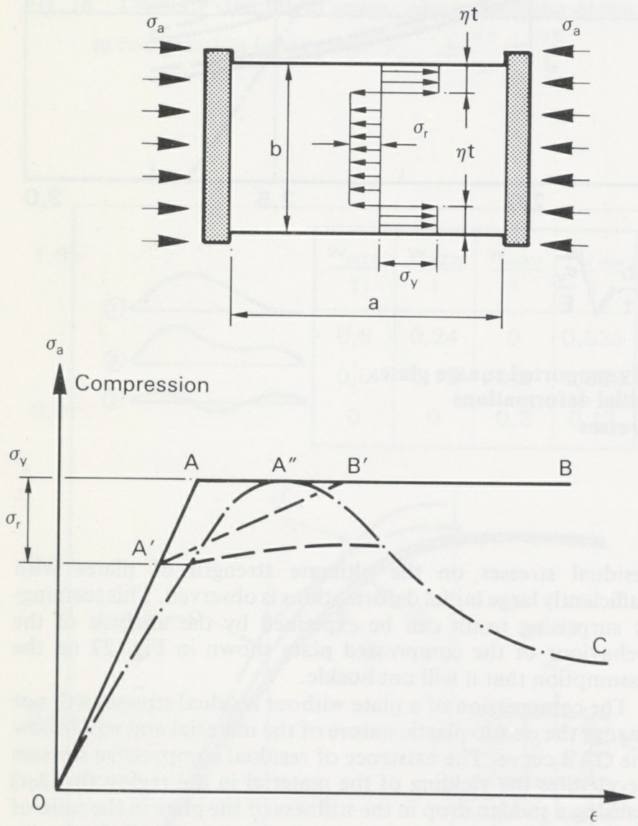


Fig. 22 Influence of residual stresses on behaviour of a flat plate which does not buckle

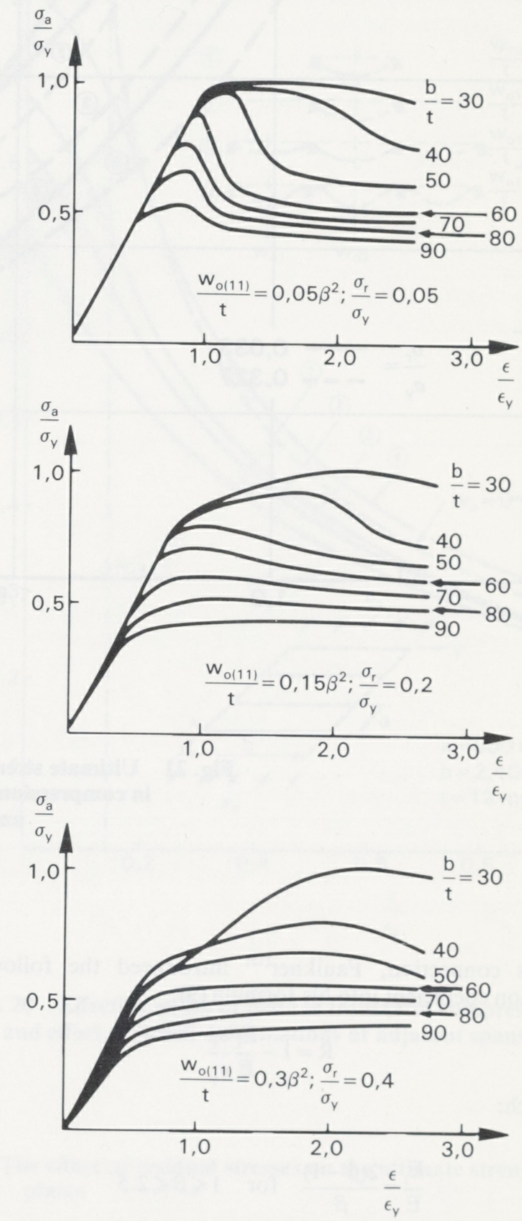


Fig. 23 Mean stress-strain curves for simply supported square plates having residual stresses and distortions

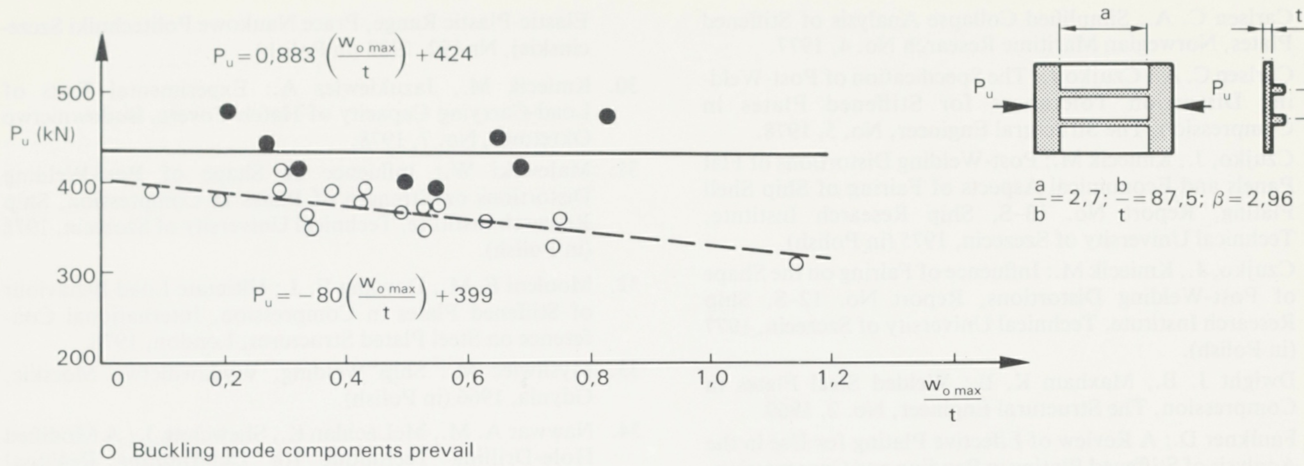


Fig. 24 Ultimate loading of longitudinally stiffened panels in compression: test results

5.4 Experimental results

Fig. 24 gives the values of the ultimate loads of twenty-five welded panels whose parameters were as following:

$a/b = 2.7$
$b/t = 87.5$
$\beta = 2.96$
$t = 4$ mm

The tests were carried out by Borzecki[5] in collaboration with the present author. Reference (5) contains details concerning the way in which the panels were made and the course of the tests. The maximum initial distortions varied between 0.17 and 1.146 of the plate thickness. Fig. 24 shows the regression curve for the ultimate loads of eight panels whose central plate had a distortion approximating one half-wave. It also shows the regression curve for the remaining seventeen panels where deformation in the form of two and three half-waves dominated.

From the data in Fig. 24 it follows that the ultimate loads P_u of the panels does not really decrease as the magnitude of the initial distortion increases if the distortion approximates one half-wave. It clearly falls, however, if other harmonics dominate in this distortion.

6. CONCLUSIONS

The results of the analytic investigations of the effect of the geometry of the initial distortion on the strength of axially compressed plates were first published many years ago.[24] Although for computational reasons they did not take residual stresses into account and were limited to the elastic range and to simple distortion forms, they showed that the geometry of the distortion has greater significance for the estimation of the influence of the distortion on the strength of the plate than its magnitude. The later analytical[31] and numerical[8, 10, 18, 40] and experimental[5] investigations discussed in this paper have confirmed the conclusions contained in[24] which one might summarise as follows:

1. The use of the maximum value of initial distortions $w_{o\,max}$ or $w_{o\,max}/t$ as a criterion for qualifying plates of ship shell plating for straightening is incorrect from the point of view of plate strength, since it is not the size of the initial distortion but its geometrical form which is decisive in respect of plate strength. Those deformation components which correspond, or are close, to the buckling mode of the plate have an unfavourable effect on strength, whereas other components have a positive effect by reducing the influence of "buckling" components. Fairing gives, in effect, a random shape generally causing an increase in the proportion of components corresponding to the buckling mode in the whole deformation, and thereby reducing plate strength despite the reduction in the maximum deformation.
2. Because it is in fact impracticable to carry out controlled fairing in production conditions which would permit the correct shape to be given to the faired plate, fairing should be kept to a minimum and only used when excessive deformation visibly affects the aesthetics of a ship or worsens its operational properties.

7. REFERENCES

1. Admiralty Ship Welding Committee Report, HMSO, London, R1(1946), R2(1948), R6(1952), R7(1952), R12(1954), R13(1954).
2. Antoniou A. *et al.*: Fabrication and Service Factors, Proceedings of the Seventh International Ship Structures Congress, Vol. I, Paris 1979.
3. Basar N. S., Stanley R. F.: Survey of Structural Tolerances in the United States Commercial Shipbuilding Industry, SSC-273, 1978.
4. Becker M., Goldman R., Pazerycki J. *et al.*: Compressive Strength of Ship Hull Girders, Part 1; Unstiffened Plates, Ship Structures Committee, Report 217, 1970.
5. Borzecki I.: Influence of Fabrication Distortions on Strength of Longitudinally Stiffened Axially Loaded Thin Plates, Report No. 56, Ship Research Institute, Technical University of Gdańsk, 1977 (in Polish).
6. Brandt A. M.: Methods of Analysis and Measurements of Residual Stresses in Structural Elements. PWN, Warszawa, 1978 (in Polish).
7. Butnicki S.: On the Possibility of Determination of Safe Service Temperature of Welded Ship Hulls by Means of Impact Tests, Technical University of Gdańsk, 1962 (in Polish).
8. Carlsen C. A.: Collapse of Stiffened Panels in Compression, Report 1. Plate Between Stiffeners, Det norske Veritas, Report No. 76-306, 1976.

9. Carlsen C. A.: Simplified Collapse Analysis of Stiffened Plates, Norwegian Maritime Research No. 4, 1977.
10. Carlsen C. A.; Czujko J.: The Specification of Post-Welding Distortion Tolerances for Stiffened Plates in Compression, The Structural Engineer, No. 5, 1978.
11. Czujko, J., Kmiecik M.: Post-Welding Distortions of Flat Panels and Economical Aspects of Fairing of Ship Shell Plating, Report No. 13–S, Ship Research Institute, Technical University of Szczecin, 1975 (in Polish).
12. Czujko, J., Kmiecik M.: Influence of Fairing on the Shape of Post-Welding Distortions, Report No. 12–S, Ship Research Institute, Technical University of Szczecin, 1977 (in Polish).
13. Dwight J. B., Maxham K. E.: Welded Steel Plates in Compression, The Structural Engineer, No. 2, 1969.
14. Faulkner D.: A Review of Effective Plating for Use in the Analysis of Stiffened Plating in Bending and Compression, Journal of Ship Research, Vol. 19, No. 1, March 1975.
15. Faulkner D.: Design Against Collapse for Marine Structures, International Symposium on Advances in Marine Technology, Trondheim, 1979.
16. Frankland J. M.: The Strength of Ship Plating under Edge Compression, USEMB Report 469, 1940.
17. Frieze P. A., Dowling P. J., Hobbes R. E.: The Measurements of Residual Stresses in Welded Ship Structures and their Effect on the Strength of Structures, Proceedings of the International Conference organized by the Institution of Civil Engineers, London, February 1973.
18. Frieze P. A., Dowling P. J., Hobbes R. E.: Ultimate Load Behaviour of Plates in Compression. International Conference on Steel Plated Structures, London, 1976.
19. Fujita Y.: Local Structural Response, International Ship Structures Congress, Hamburg, 1973.
20. Fujita Y., Kalinin V. S., Kavlie D., Kmiecik M. *et al.*: Non-Linear Structural Response, International Ship Structures Congress, Boston, 1976.
21. Kmiecik M.: Influence of Initial Deformations on Strength of Rectangular Plates in Compression. Technical University of Gdańsk, 1970 (in Polish).
22. Kmiecik M.: The Load-Carrying Capacity of Axially Loaded Longitudinally Stiffened Plate Panels having Initial Deformations, SFI Trondheim, 1970.
23. Kmiecik M.: Influence of Shape of Initial Deformations on Strength of Plates in Compression, Budownictwo Okretowe, No. 5 and No. 6, 1971.
24. Kmiecik M.: Behaviour of Axially Loaded Simply Supported Long Rectangular Plates Having Initial Deformations, SFI Trondheim, 1971.
25. Kmiecik M.: Model Tests of Load-Carrying Capacity of Longitudinally Stiffened Plates in Compression, Zeszyt No. 1, 1972 (in Polish).
26. Kmiecik M.: Are the Post-Welding Distortions Detrimental? Budownictwo Okretowe, No. 8, 1974 (in Polish).
27. Kmiecik M.: Are the Post-Welding Distortions Detrimental? Proceedings of International Ship Structures Congress, Hamburg, 1974.
28. Kmiecik M., Jazukiewicz A.: Experimental Tests of Load-Carrying Capacity of Hatch Covers, VII Krajowa Konferencja Wytrzymalości i Badania Materialów, Gdańsk, 1977 (in Polish).
29. Kmiecik M.: Incremental Equations of Non-Linear Theory of Plates with Initial Deflections and Stresses in Elastic-Plastic Range, Prace Naukowe Politechniki Szczecinskiej, Nr 192, 1982 (in Polish).
30. Kmiecik M., Jazukiewicz A.: Experimental Tests of Load-Carrying Capacity of Hatch Covers, Budownictwo Okretowe, No. 7, 1978.
31. Majewski W.: Influence of Shape of Post-Welding Distortions on Strength of Plates in Compression, Ship Research Institute, Technical University of Szczecin, 1978 (in Polish).
32. Mooleni F. M., Dowling P. J.: Ultimate Load Behaviour of Stiffened Plates in Compression, International Conference on Steel Plated Structures, London, 1976.
33. Myśliwiec M.: Ship Welding, Wydawnictwo Morskie, Gdynia, 1966 (in Polish).
34. Nawwar A. M., McLachlan K., Shewchuk J.: A Modified Hole-Drilling Technique for Determining Residual Stresses in Thin Plates. Experimental Mechanics, No. 6, 1976.
35. Orloś Z.: Experimental Analysis of Strains and Stresses, PWN, Warszawa, 1977 (in Polish).
36. Pattes H. E., Evans R. M.: Effect of Flame and Mechanical Straightening on Material Properties of Weldments, SSC–207, 1970.
37. Pfau H., Kmiecik M., Knappe H. J.: Ein Beitrag zur Berechnung der Verformungen und Spannungen in druckbelasteten isotropen Rechteckplatten grosser Ausbiegung unter Verwendung von geeignetem Differenzenmethoden, Schiffbauforschung 5/6, 1978.
38. Schiffbau-Kolloquium Werftindustrie-Germanischer Lloyd, Schiff und Hafen, 5/1977.
39. Somerville W. L., Swan J. W., Clark J. D.: Measurements of Residual Stresses and Distortions in Stiffened Panels, Journal of Strain Analysis, No. 2, 1977.
40. Smith C. S., Bäcklund J., Bergan P. G., Fujita Y., Jones N., Kmiecik M. *et al.*: Non-linear Structural Response, Proceedings of the Seventh International Ship Structures Congress, Vol. I, Paris, 1979.
41. Smith C. S.: Compressive Strength of Welded Steel Ship Grillages, The Royal Institution of Naval Architects, 1975.
42. Ueda Y., Fukuda K., Nakacho K.: Basic Procedures in Analysis and Measurements of Welding Residual Stresses by the Finite Element Method, Welding Institute, London, 1977.
43. Ueda Y., Yasukawa W., Ikegami H., Ohminami R.: Effects of Welding Residual Stresses and Initial Deflection on Rigidity and Strength of Square Plates Subjected to Compression, Transactions of Japanese Welding Research Institute, No. 2, 1975 and No. 1, 1977.
44. Welding Handbook, American Welding Society, Vol. I, 1976.
45. Wilken K.: The Measurements of Residual Stresses in Welded Ship Structures and their Effect on the Strength of Structures.
46. Winokurow W.: Post-Welding Deformations and Stresses, Moscow, 1968 (in Russian).
47. Volmir S.: Flexible Plates and Shells, Moscow, 1956 (in Russian).
48. Young B. W., Elliott P., Bowers G.: Residual Stresses and Measurements of Tolerances, Proceedings of the International Conference organized by the Institution of Civil Engineers, London, February, 1973.

9. Carlsen C. A.: Simplified Collapse Analysis of Stiffened Plates, Norwegian Maritime Research No. 4, 1977.
10. Carlsen C. & Czujko J.: The Specification of Post-Welding Distortion Tolerances for Stiffened Plates in Compression, The Structural Engineer, No. 5, 1978.
11. Czujko J., Kmiecik M.: Post-Welding Distortions of Flat Panels and Economical Aspects of Fairing of Ship Shell Plating, Report No. 19-S, Ship Research Institute, Technical University of Szczecin, 1975 (in Polish).
12. Czujko J., Kmiecik M.: Influence of Fairing on the Shape of Post-Welding Distortions, Report No. 12-S, Ship Research Institute, Technical University of Szczecin, 1977 (in Polish).
13. Dwight J. B., Moxham K. E.: Welded Steel Plates in Compression, The Structural Engineer, No. 2, 1969.
14. Faulkner D.: A Review of Effective Plating for Use in the Analysis of Stiffened Plating in Bending and Compression, Journal of Ship Research, Vol. 19, No. 1, March 1975.
15. Faulkner D.: Design Against Collapse for Marine Structures, International Symposium on Advances in Marine Technology, Trondheim, 1979.
16. Frankland J. M.: The Strength of Ship Plating under Edge Compression, USEMB Report 469, 1940.
17. Frieze P. A., Dowling P. J., Hobbs R. E.: The Measurements of Residual Stresses in Welded Ship Structures and their Effect on the Strength of Structures, Proceedings of the International Conference organized by the Institution of Civil Engineers, London, February 1977.
18. Frieze P. A., Dowling P. J., Hobbs R. E.: Ultimate Load Behaviour of Plates in Compression, International Conference on Steel Plated Structures, London, 1976.
19. Fujita Y.: [illegible], International Ship Structures Congress, [illegible], 1973.
20. Fujita Y., [illegible], Kmiecik M. et al.: Non-Linear Structural Response, International Ship Structures Congress, Boston, 1976.
21. Kmiecik M.: Influence of Initial Deformations on Strength of Rectangular Plates in Compression, Technical University of Gdańsk, 1970 (in Polish).
22. Kmiecik M.: The Load-Carrying Capacity of Axially Loaded Longitudinally Stiffened Plate Panels having Initial Deformations, SIT Trondheim, 1970.
23. Kmiecik M.: Influence of Shape of Initial Deformations on Strength of Plates in Compression, Budownictwo Okrętowe, No. 6, 1971.
24. Kmiecik M.: Behaviour of Axially Loaded Simply Supported Long Rectangular Plates Having Initial Deformations, SIT Trondheim, 1971.
25. Kmiecik M.: Model Tests of Load-Carrying Capacity of Longitudinally Stiffened Plates in Compression, Zeszyt No. 1, 1972 (in Polish).
26. Kmiecik M.: Are the Post-Welding Distortions Dangerous? Budownictwo Okrętowe, No. 8, 1974 (in Polish).
27. Kmiecik M.: Are the Post-Welding Distortions Dangerous? Proceedings of International Ship Structures Congress, Hamburg, 1976.
28. Kmiecik M., [illegible] A.: Experimental Tests of Load-Carrying Capacity of Hatch Covers, [illegible], Gdańsk, 1977 (in Polish).
29. Kmiecik M.: Incremental Equations of Non-Linear Theory of Plates with Initial Deflections and Stresses in Elastic-Plastic Range, Prace Naukowe Politechniki Szczecińskiej, Nr 192, 1982 (in Polish).
30. Kmiecik M., Jakubowicz A.: Experimental Tests of Load-Carrying Capacity of Hatch Covers, Budownictwo Okrętowe, No. 7, 1978.
31. Majewski W.: Influence of Shape of Post-Welding Distortions on Strength of Plates in Compression, Ship Research Institute, Technical University of Szczecin, 1978 (in Polish).
32. Moolani F. M., Dowling P. J.: Ultimate Load Behaviour of Stiffened Plates in Compression, International Conference on Steel Plated Structures, London, 1976.
33. Myśliwiec M.: Ship Welding, Wydawnictwo Morskie, Gdynia, 1960 (in Polish).
34. Nawwar A. M., McLachlan K., Shewchuk J.: A Modified Hole-Drilling Technique for Determining Residual Stresses in Thin Plates, Experimental Mechanics, No. 6, 1976.
35. Orłoś Z.: Experimental Analysis of Strains and Stresses, PWN, Warszawa, 1977 (in Polish).
36. Pattee F., Evans R. M.: Effect of Flame and Mechanical Straightening on Material Properties of Weldments, SSC-207, 1970.
37. Pfau H., Kmiecik M., Knappe H. J.: Ein Beitrag zur Berechnung der Verformungen und Spannungen in druckbelasteten isotropen Rechteckplatten grosser Ausbiegung unter Verwendung von geeigneten Differenzenmethoden, Schiffbauforschung 3-4, 1978.
38. [illegible], Schiff und Hafen, [illegible].
39. Somerville W. L., Swan J. W., Clarke J. D.: Measurements of Residual Stresses and Distortions in Stiffened Panels, Journal of Strain Analysis, No. 2, 1977.
40. Smith C. S., [illegible] J., Bergan P. G., Fujita Y., [illegible] N., Kmiecik M. et al.: Non-linear Structural Response, Proceedings of the Seventh International Ship Structures Congress, Vol. 1, Paris, 1979.
41. Smith C. S.: Compressive Strength of Welded Steel Ship Grillages, The Royal Institution of Naval Architects, 1975.
42. Ueda Y., Fukuda K., Nakacho K.: Basic Procedures in Analysis and Measurements of Welding Residual Stresses by the Finite Element Method, Welding Institute, London, 1977.
43. Ueda Y., Yasukawa W., [illegible] H., Ohminami R.: Effect of Welding Residual Stresses and Initial Deflection on Rigidity and Strength of Square Plates Subjected to Compression, Transactions of Japanese Welding Research Institute, No. 2, 1975 and No. 1, 1977.
44. Welding Handbook, American Welding Society, Vol. 1, [illegible].
45. Wilson R.: The Measurements of Residual Stresses in Welded Ship Structures and their Effect on the Strength of Structures.
46. Winokurow W.: Post-Welding Deformations and Stresses, Moscow, 1968 (in Russian).
47. Volmir S.: Flexible Plates and Shells, Moscow, 1956 (in Russian).
48. Young R. W., [illegible], Barnes G.: Residual Stresses and Measurements of Tolerances, Proceedings of the International Conference organized by the Institution of Civil Engineers, London, February, 1977.

a) Recommended deformation correct. methods?

b) Why two identical ships have different deformation

c) Academic v practical solutions.

d) Fitting of fairing bar to improve strength of faired plate. gets around any ~~doubts as to~~ doubts as to the final strength of the plate.

Tony Viner
Geoff Cole
Reed

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Discussion

on the paper

A REVIEW OF FABRICATION DISTORTION TOLERANCES FOR SHIP PLATING IN THE LIGHT OF THE COMPRESSIVE STRENGTH OF PLATES

by

Professor Marian Kmiecik
(Guest Lecturer)

Paper No. 6. Session 1986–87

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by

Professor Marian Kmiecik

Head of Ship Structure Mechanics Department,

Technical University of Szczecin

DISCUSSION

From Mr. F. Reid:

Professor Kmiecik's paper is an interesting one and he deserves our thanks for coming to deliver it.

Of the five listed factors which may determine the magnitude of permissible fabrication distortions, I am not sure that the effect on strength is the major consideration. I do not know of any case of a major ship casualty attributable to fabrication distortions.

The effect of distortion in the bottom forward pounding area and in panting regions could be considered more detrimental than in some other locations, as these deflections are likely to increase in service. We have had several cases of bottom pounding damage in fast reefer ships, for example.

The modern concept of ship structure design, whilst good from the point of view of increased deadweight, does not help the fabricator faced with welding distortion. Today's ships, compared to their predecessors of 10 years ago, have much lighter scantlings, particularly in the shell envelope plating. A 260m long tanker built 10 years ago had bottom plating of about 25mm, today the thickness would be about 18mm. Welding distortions which used to be more pronounced in way of plating at ends and superstructures are now found throughout the structure.

Acknowledging Professor Kmiecik's conclusions that the geometry of the distortion has greater significance on the strength than magnitude and because it is impractical to carry out controlled fairing in production, and that fairing should be kept to a minimum, one must ask what is to be considered permissible distortion – are presently accepted standards sufficient?

I hope Professor Kmiecik is going to give us another paper entitled "How to minimise fabrication distortion" as there is no doubt prevention is the best cure.

From Mr. A. C. Viner:

Professor Kmiecik is an acknowledged expert on the strength of plated structures and we are fortunate to have this excellent paper from him. It will make an impressive addition to the Association's transactions.

In the paper Professor Kmiecik has demonstrated some of his pioneering work on the effects of different shapes of initial deformation on the strength of plating under uni-axial compression. He comes to the conclusion that fairing should be kept to a minimum and only used when excessive deformation visibly affects the aesthetics of a ship or worsens its operational properties.

I hope that this is not intended to imply that, from the structural aspect, we need not set standards of flatness for ship plating. Initial deformations of plating are not as harmful to structural strength as unfairness of stiffening members, but there are still good reasons for keeping them in check; namely:

(i) From the ultimate strength aspect, there are few parts of the ship structure which have to withstand only uni-axial compression. In particular, transverse compression induced mainly by bending of transverses can result in a serious reduction of buckling strength, which will be aggravated by initial deformations. In bi-axial compression the maximum deformation is normally more significant than the ripple component.

(ii) In Rule formulations and direct calculation there is assumed either implicitly or explicitly a certain effectiveness of plating which contributes to the flexural behaviour of stiffening members or of the hull girder itself. Loss of plate effectiveness can occur at low stress levels as a result of initial unfairness and this applies to tension as well as compression. Such loss of effectiveness will lead to higher stresses in connections and consequent reduction of fatigue life. We must be sure that the ship structure is built to the standard of fairness which is assumed in design appraisal.

(iii) Finally, a standard of fairness provides a control over workmanship procedures. Insistence on the correction of large deformations provides an incentive to develop the best construction procedures.

Lloyd's Register's recommended ship construction tolerances and defect correction procedures are given in Hull Structures Report No. 84/38. This document is primarily related to a Quality Assurance Scheme but, when a Scheme is not in operation at a particular shipyard, the standards may be used by Surveyors for guidance to complement their experience and judgement. The plate deformation tolerances recommended at present for new construction are as follows:

ITEM	*STANDARD*	*LIMIT*
Strength deck, shell plating, webs of primary members, all within 0.6L amidships	$w_o = \dfrac{b}{200\sqrt{k}}$	$w_o = \dfrac{b}{133\sqrt{k}}$
All other plating	$w_o = \dfrac{b}{120\sqrt{k}}$	$w_o = \dfrac{b}{80\sqrt{k}}$

where w_o = maximum plate deformation between adjacent stiffeners

k = higher tensile steel factor (i.e. ratio of yield stresses of mild steel and higher tensile steel with lower limit 0.72).

Panels with deformations exceeding the permissible values are to be faired by local heating. When flame straightening is employed for fairing purposes the local heating of the steel should not exceed a temperature of 900°C, i.e. a visible red heat.

In order to reduce the time involved in the cooling process, water cooling is permitted provided the temperature of the heating zone is allowed to cool back to below 600 C, i.e. a black heat, before water is applied. It is essential to ensure that the above-stated temperatures are not exceeded and, in this respect, the heating and cooling operations carried out in the fairing of deformed plate materials are to be strictly controlled. The 'standard' values are intended to apply to 95% of plating in a particular area. Not more than 5% of plate panels are permitted to have deformations greater than the standard and correction is necessary for deformations exceeding the 'limit' values. For short superstructures and deckhouses only the limit values need be complied with.

The overriding consideration in formulating these recommendations has been to make them as simple to apply as possible, because a Surveyor does not have time to take extensive measurements. The levels of the recommended initial deformations have been derived partly from the strength and stiffness considerations discussed above and partly from experience of what can be achieved in practice. It may be noted that the values are comparable to those in the Japanese Standards (JSQS). The standard range for the main hull lies above Antoniou's mean values (49) given by

$$\frac{w_{o\,max}}{t} = 0.00022\left(\frac{b}{t}\right)^{1.69}$$

based on measurements on plate panels with breadth/thickness ratios from 22 to 77. The limit values are of the same order as those given by Antoniou's formulation for mean + 2 standard deviations. It would be useful if the Author could indicate a similar expression for the mean + 2 standard deviations of the Polish measurements.

The setting of standards is an evolving process and the above values could be revised in the light of the latest advances in knowledge of stiffened plate behaviour including that given in this paper. I should be grateful for the Author's comments on the recommended plate deformation tolerances and would also welcome a more detailed explanation of the elimination of the dependency on yield stress as proposed on page 4 of the paper.

REFERENCE

49 ANTONIOU A.C.: On the Maximum Deflection of Plating in Newly Built Ships, Journal of Ship Research, March 1980.

From Mr. G. H. Sole

Dr. Kmiecik has presented an interesting paper, illustrating that it is the initial deformations in the short wave lengths that lead to the largest reductions in plate strength when that plate is subjected to an axial load. This conclusion is similar to that reached by other authors in recent papers. There are however some differences in that for example in reference (50) it is considered that dividing the panel shape into its Fourier components tends to result in an averaging process which reduces the influence of local indents. Another recent paper (51) finds that it is the magnitude of components in a wave length slightly shorter than that for the elastic buckling mode which cause the greatest loss of panel longitudinal strength. It appears from the findings of these two papers, and from Dr. Kmiecik, that probably the most important parameter is the local curvature of the initially deformed panel in the direction of the stress. To express this more simply, the larger the curvature, the bigger the indentation, the weaker the panel.

The above comments refer to one isolated panel loaded purely in axial compression, and take no account of the effect of adjacent panels, stiffeners, and other load components. In order to obtain some understanding of the behaviour of a stiffened panel subjected to both longitudinal and transverse compression, a three-dimensional finite element model of a typical structural configuration was constructed. This model consisted of an inverted angle, attached to plating of aspect ratio three and breadth to thickness ratio equal to sixty, and supported along the transverse frame line. The area of analysis extended over four adjacent quarter panels, and computations were performed using the non-linear finite element program MARC, including large deflections and plasticity.

All material was taken to be of normal strength structural steel. Symmetrical boundary conditions along the panel centrelines were assumed for rotations, and all four edges were constrained to remain straight during loading, which was by displacement control. Initial imperfections included an overall deformation of the inverted angle and plate in the longitudinal direction, combined with local panel dishing in the one half wave shape. In addition small values of the buckling mode shapes were included to ensure that the lowest failure mode was triggered. A residual stress distribution caused by fillet welding of the stiffener and the frame to the plate was included in the model. The magnitude in the compression zones of the panel (i.e. away from the weld lines) amounted to 10% of the yield stress in the longitudinal direction, and 4% transversely.

The objective of this work was to determine the ultimate failure loads of the structure under various combinations of bi-axial compression, and these results can be broadly summarised by the interaction diagram in Figure D1 where the applied mean stresses are normalised by the yield value. These curves were obtained on the basis of four different criteria, and the interested reader is referred to reference (52) for details. Essentially, however the curves can be divided into two components with a knee point at approximately a transverse stress ratio (σ/σ_{yield}) of 0.15. The lower vertical part (A to B) is then characterised predominantly by a collapse in the three half-waves buckling mode, whilst the upper part (B to C and onwards) shows a failure mainly in the long wave form (see Figure D2)

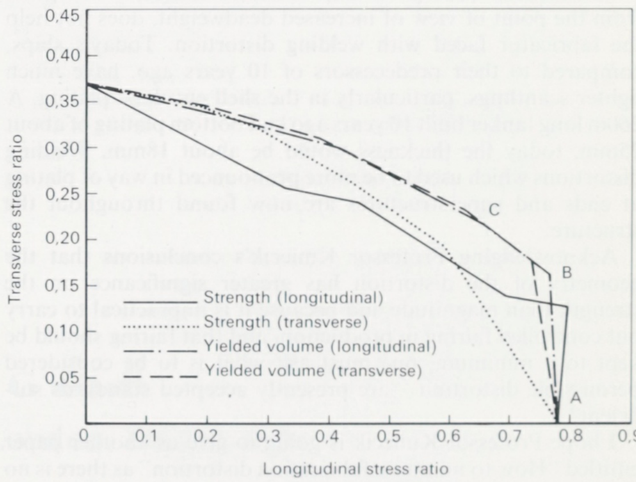

Figure D1 Interaction curves

The essential point behind this discussion is that the ultimate strength of panels loaded by combinations of stress will be sensitive to initial deformations of varying shapes. Therefore, in an area of the ship subjected primarily to axial compression, for example the upper deck in the midships region, the presence of initial imperfections characterised by short wave lengths (i.e. large curvatures) should be avoided, and it is agreed that fairing could diminish plate strength if it results in this type of shape. However, in other areas of the hull girder where panels are subjected to combined loads, other mode components including the one half wave shape should be taken into consideration, and the above example appears to imply that fairing may be beneficial, provided that this is performed properly.

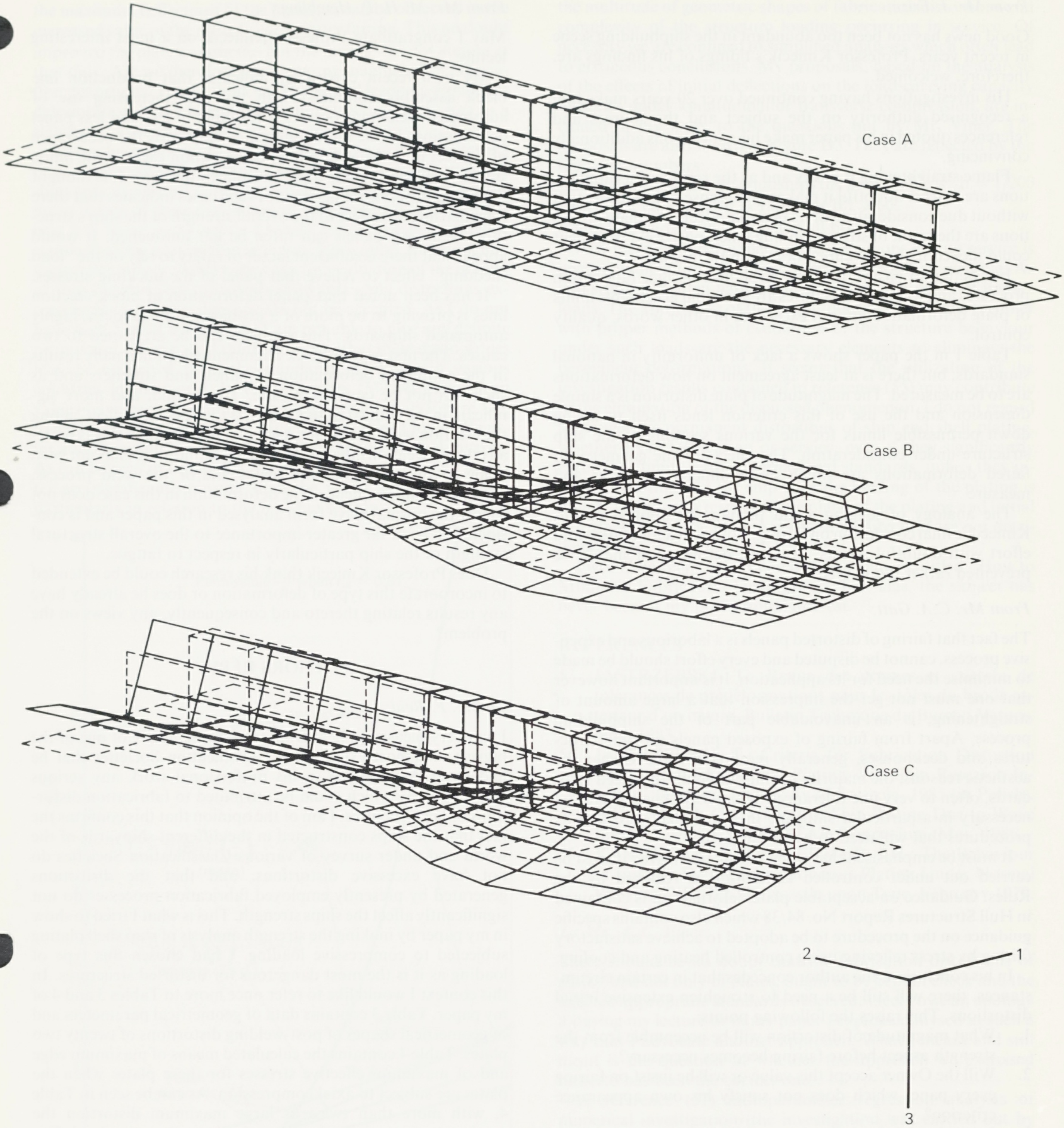

Figure D2 Deformed Shapes in Axial and Bi-Axial Compression

REFERENCES

50 DOW, R.S. and SMITH, G.S.: Effects of the Localised Imperfections on Compressive Strength of Long Rectangular Plates, J. of Constructional Steel Research, Vol. 4 No. 1 1984.

51 UEDA, Y. and YAO, T.: The Influence of Complex Initial Deflection Modes on the Behaviour and Ultimate Strength of Rectangular Plates in Compression, J. of Constructional Steel Research, Vol. 5 No. 4 1985.

52 VINER, A.C.: Development of Ship Strength Formulations, Advances in Marine Structures, ed. C. S. Smith and J. D. Clarke, Elsevier Applied Science Publishers, 1987.

From Mr. J. Frize:

Good news has not been too abundant in the shipbuilding scene in recent years. Professor Kmiecik's tidings of his findings are, therefore, welcomed.

His investigations having continued over 20 years make him a recognised authority on the subject and the sources and references quoted in his paper make his conclusions additionally convincing.

Flame straightening is costly and as the results of its applications are at least doubtful it makes good sense not to resort to it without due consideration of its possible effects. As plate distortions are the result of stresses finding relief, post-weld treatment could indeed make the cure worse than the disease.

Far better to ensure that the structural design, fabrication processes and welding procedures are such that acceptable limits of plate deformation are not exceeded; in other words, quality control.

Table 1 in the paper shows a lack of uniformity in national standards; but there is at least agreement on how deformations are to be measured. The magnitude of plate distortion is a simple dimension and the use of this criterion lends itself to laying down permissible limits for the various sections of the ship structure under consideration. The effects of the geometry of faired deformations are much more difficult to assess and measure.

The analogy of jogging could perhaps be applied to Dr. Kmiecik's final conclusion on fairing. Both cost a lot of time and effort with doubtful results: extensive deformation should be prevented rather than cured.

From Mr. C.A. Gatt:

The fact that fairing of distorted panels is a laborious and expensive process, cannot be disputed and every effort should be made to minimise the need for its application. It is important however that one must not get the impression that a large amount of straightening, is an unavoidable part of the shipbuilding process. Apart from fairing of exposed panels on superstructures and deckhouses, generally required by the Owner for aesthetic reasons, the majority of ships are built to agreed standards, often to very fine tolerances. If extensive fairing is found necessary in a particular shipyard then it is that yard and its procedures that will need to be investigated.

It must be emphasised that when fairing is involved it must be carried out under controlled conditions as required by the Rules. Guidance on acceptable plate deformation is contained in Hull Structures Report No. 84/38 which also contains specific guidance on the procedure to be adopted to achieve satisfactory fairing by stress relieving using controlled heating and cooling.

In his conclusion, the author concedes that in certain circumstances, there will still be a need to straighten extensive initial distortions. This raises the following points:

1. What magnitude of distortion will be acceptable from the strength aspect before fairing becomes necessary?
2. Will the Owner accept this value or will he insist on fairing every panel which does not satisfy his own appearance criterion?
3. It is acknowledged that fairing of deeper distortions may result in more pronounced buckling mode deformations and for this reason this practice must be kept to a minimum. In view of the difficulty encountered in the fairing operation of deep distortions, shipyards often resort to cropping and part renewing of plates, panel stiffening, or a combination of both these methods.

Finally it must also be stated that there is no evidence to suggest that failure has occurred as a direct result of fairing distorted panels and any deviation from this long established practice must be very carefully approached. Perhaps greater emphasis should be placed on researching and promoting ship construction equipment and processes which minimise initial panel deformations.

From Mr. M. H. P. Hembling:

May I congratulate Professor Kmiecik on a most interesting lecture.

My own recent experience indicates that production line block assembly combined with the ever increasing use of automated welding processes is resulting in less and less panel deformation. In the more automated shipyards the occurrence of a panel in mid-block, whose deformation exceeds the limits shown in Table 1, tends to be an isolated incident. It is thought that since in-service experience of such cases indicates that there is no detrimental effect to the overall strength of the ship's structure, then such panels can often be left untouched. It would appear that there is sufficient factor of safety to rely on the "load shedding" effect to relieve that panel of the buckling stresses.

It has been noted that panel deformation at block erection lines is proving to be more of a problem in the modern, highly automated shipyards. This problem can be attributed to two causes. The first is poor block alignment which normally results in the combined deformation of panels and stiffeners and as such will not be considered here. The second, and more significant in the context of this paper, is the "peaking effect" along the erection weld line within each individual panel; this is basically due to weld shrinkage, for which due allowance has not been made in the setting of the plates and/or the weld process, parameters and sequence. The deformation in this case does not follow the normal wave form analysed in this paper and is considered to be of far greater importance to the overall structural strength of the ship particularly in respect to fatigue.

Does Professor Kmiecik think his research could be extended to incorporate this type of deformation or does he already have any results relating thereto and consequently, any views on the problem?

AUTHORS REPLY

To Mr. F. Reid:

I was very pleased to hear from the representative of one of the biggest and most experienced Classification Societies that he had never encountered in his professional work any serious ship's casualty which could be attributed to fabrication distortions of her structures. I am of the opinion that this confirms the fact that the ships constructed in the different shipyards of the world and under survey of various Classification Societies do not have excessive distortions, and that the distortions generated by presently employed fabrication processes do not significantly affect the ships strength. This is what I tried to show in my paper by making the strength analysis of ship shell plating subjected to compressive loading. I had chosen this type of loading as it is the most dangerous for distorted structures. In this context I would like to refer once more to Tables 3 and 4 of my paper. Table 3 contains data of geometrical parameters and of geometrical shapes of post-welding distortions of twenty two plates. Table 4 contains the calculated means of maximum edge and of maximum effective stresses for these plates when the plates are subject to axial compression. As can be seen in Table 4, with more than twice as large maximum distortion the maximum stresses remained practically at the same level. For me it is a clear indication that, with the existing geometryof fabrication distortions, we ascribe too much significance to the influence of these distortions on ship structures strength.

Our efforts to reduce excessive distortions do not bring the expected effects either. This can be seen in Figure 14 in my paper. The plate in which, after fairing, the maximum distortion had been reduced by more than 50 per cent, showed the highest maximum effective stresses. This tendency is also confirmed by the statistical data given in Table 2. The faired rectangular plates had practically the same value of the ratio of buckling mode component w_{ob} to plate thickness t as before fairing. Thus the most detrimental component was not eliminated during fairing. The only positive effect of that undertaking was a reduction of

the maximum deflections of the plates, as can be observed in the increased value of the ratio $\bar{w}_{ob}/\bar{w}_{omax}$ after fairing. This no doubt improved the plate's aesthetics but the fairing was not aiming at that only. Are we therefore to accept any magnitude of fabrication distortions? I don't think so. This would lead to a significant deterioration in the execution of ship structures and thus to a significant lowering of their quality and aesthetics.

Two solutions occur to me here. The first based on the results of measurements of maximum distortions in a large number of shipyards in various countries would allow calculation of the mean values of these distortions as a function of plate slenderness b/t. These mean values I would call the mean worldwide standard of the present ship structure fabrication. Subsequently these mean values could be the basis for evaluation of the quality of ship construction in various shipyards. Any distortions exceeding these mean values should not be accepted. As I do not have such data at my disposal I am not able to give any definite proposal in this matter. I think, however, that these mean values may be estimated from the standards employed in various countries in the world. In the references (53–54) these mean values have been determined and they are given in Figure D3, together with the permissible values specified by Lloyd's Register and the results of measurements taken in Polish shipyards. As can be seen in Figure D3 the requirements of Lloyd's Register are significantly severer than the world average standard particularly in relation to the midship section.

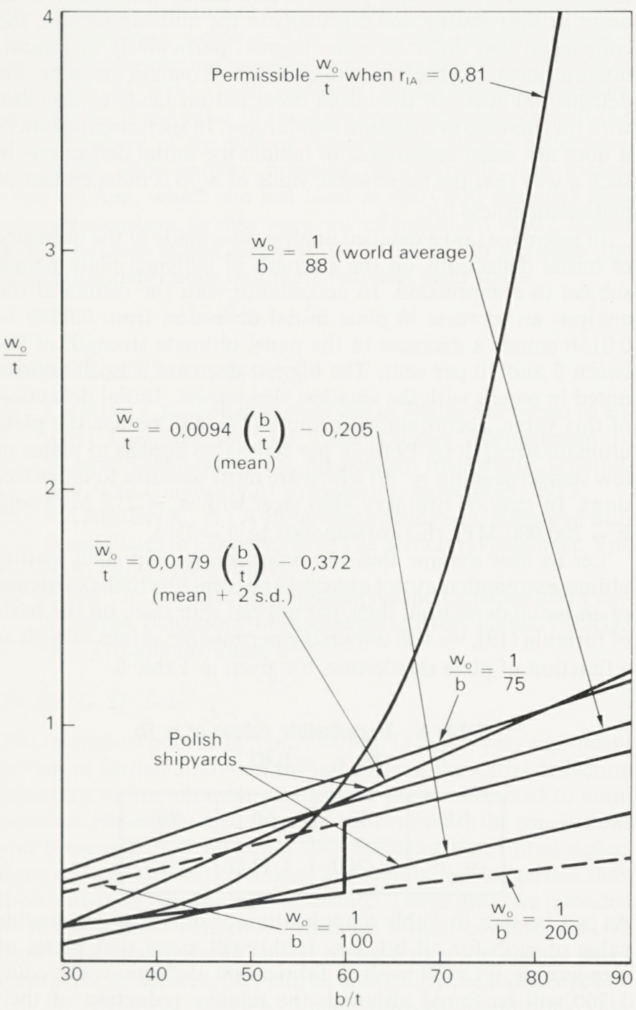

Figure D3 Values of Permissible and Measured Post Welding Distortions of Plates

The other approach which I find more rational consists in basing the permissible fabrication distortion standards on an appropriate analysis of influence of distortions on structure strength. This is, however, a very difficult task to perform due to the multitude of geometric shapes of fabrication distortions and complexity of the structure loading occurring in service. Of necessity we are confined to simplified analyses which often lead to erroneous conclusions. My proposals, based on the analysis of the effects of initial deflections on the load-carrying capacity of plates under longitudinal or transverse compression, are included in my reply to the questions put by Mr. Viner. The proposals are also shown in Figure D3. They are reflected by the following numbers:

– all plating within 0.6L amidships the permissible $w_o/b = 1/200$ for $b/t < 60$ and $w_o/b = 1/100$ for $b/t \geqslant 60$,
– ship hull ends $w_o/b = 1/75$ for all b/t.

Any increase in permanent plate deflections during service is unacceptable and should be corrected as soon as possible. As you stressed it often happens in ship ends as a result of impact loads. The knowledge of the magnitudes of these loads together with proper methods of estimation of the structure behaviour under such loads are the necessary elements to eliminate the permanent deformations. It seems that the utilization of the investigation results contained in reference (55) may contribute to the elimination of the discussed phenomenon. I do not regard as dangerous permanent distortions of ship end shell plating, whether from fabrication or service, even if they significantly exceed the limits set by international standards, unless they increase with time. At the ship ends the loading of the plating is mainly lateral with respect to the plane of the plating and initial deformations of the plating in the same direction are not detrimental (56).

I would be very glad if I could write a paper entitled "How to Minimize Fabrication Distortions", but, alas, the subject has never been of my professional interest.

REFERENCES

53 JASTRZEBSKI, T.: Revue des normes et criterés de tolérances de prédéformations pour les tôles du brodé extérieur d'un navire, Bulletin Technique du Bureau Veritas, Nov. 1983.

54 JASTRZEBSKI, T.: Criteria for Fabrication Distortion Tolerances of Ship Shell Plating, The Ship Research Institute, Technical University of Szczecin, 1985 (in Polish).

55 WIERNICKI, C.: Damage of Ship Plating Due to Wave Impact Loads, SNAME STAR Symposium, May, 1986.

56 KAMTEKAR, A.G. and FIDLER, J.H.: The Behaviour of Initially Deformed Continuous Plating Under Normal Pressure, NECIES, Newcastle upon Tyne, February 1979.

To Mr. A. C. Viner:

There should be no doubt that the fabrication distortions of plates, and not only of plates, ought to be kept in check and the selected appropriate standards observed. I feel I owe an apology if during my lecture or in my paper I expressed myself in such a way that I advocate another approach. What I meant and still mean is the question of whether the approach we exercised hitherto has been the correct one.

In reference (57) multiparameter regression analysis of numerical investigation (the investigation was carried out by finite element method) has yielded the following formulae for evaluation of ultimate strength of simply supported axially compressed plates having initial deflections and stresses:

$$\phi = \frac{\sigma_{up}}{\sigma_Y} = \phi_{pu} r_{IR} r_A \qquad (17)$$

where

ϕ_{pu} – is the load-carrying capacity of a nearly perfect square plate,
r_{IR} – is the load-carrying capacity correction factor due to initial imperfections,
r_A – is the load-carrying capacity correction factor due to plate aspect ratio.

The load-carrying capacity of nearly perfect square plates with constrained unloaded edges.

$$R_{pu} = 2.74/\beta - 2.56/\beta^2 - 0.92/\beta^3 \quad (18)$$

and is valid for $\beta \geq 1$.

The correction factor due to initial imperfections (initial deflections w_o of the shape of regular half-wave and initial stresses σ_r of distribution as in Figures 4 and 5 and of magnitude in the range $\sigma_r/\sigma_y = 0.10$ to 0.30):

$$r_{IA} = 1.52[1 - 2.53(w_o/b)^{0.11}c_\beta] \quad (19)$$

where

$$c_\beta = 1.21/\beta - 1.47/\beta^2 - 0.59/\beta^3 \quad (20)$$

for $\beta \geq 1$.

The correction factor due to the plate aspect ratio

$$r_A = 1 + \beta^{0.3}(w_o/b)^{0.4}(a/b - 1)^{0.5} \quad (21)$$

for longitudinal compression and aspect ratio $a/b \leq 5$, and

$$r_A = 1 - 0.72(1 - b/a) \quad (22)$$

for transverse compression and aspect ratio $a/b \leq 8$.

Table 5 contains the calculation results obtained by use of the above formulae. The calculations were performed for square and rectangular plates the latter being subjected either to longitudinal compression (parallel to the longer edges) or transverse compression (parallel to the shorter edges).

Table 5 Ultimate strength of imperfect plates under longitudinal or transverse compression

β	$a/b = 1$	$= 2$		$= 3$		$= 4$	
	$w_{o(11)}/b = 1/200$						
	ϕ	ϕ_L	ϕ_T	ϕ_L	ϕ_T	ϕ_L	ϕ_T
1	0.893	1.000	0.571	1.044	0.464	1.078	0.411
2	0.720	0.826	0.461	0.870	0.374	0.904	0.331
3	0.635	0.741	0.406	0.785	0.330	0.819	0.292
4	0.565	0.668	0.362	0.711	0.294	0.743	0.260
	$w_{o(11)}/b = 1/133$						
1	0.857	0.978	0.549	1.028	0.446	1.067	0.394
2	0.694	0.815	0.444	0.865	0.361	0.903	0.319
3	0.618	0.739	0.396	0.790	0.321	0.828	0.284
4	0.554	0.672	0.354	0.721	0.288	0.759	0.255

ϕ – ultimate strength of compressed square plates or of rectangular ones under longitudinal compression with buckling mode initial deflection,

ϕ_L – ultimate strength of longitudinally compressed rectangular plates,

ϕ_T – ultimate strength of transversely compressed rectangular plates.

It is generally known that the ultimate strength of a square plate is equal to the ultimate strength of a rectangular one under longitudinal compression if the rectangular plate initial deflection has the shape of its buckling mode component. Therefore in this particular case it is possible to draw conclusions on the influence of initial deflections on strength of a rectangular plate on the basis of the analysis made for a square plate. In the calculations performed it was assumed that the initial plate deflections do not exceed the permissible values established by Lloyd's Register. Hence for the ordinary ship steel w_o/b was taken equal to 1/200 and to 1/133.

The calculation results given in Table 5 confirm the facts that are generally known:

1. Decrease in plate ultimate strength occurs only when the shape of the plate initial deflection is the same as the shape of the plate buckling mode deflection. The same applies to any structure, not only to plates.
2. The detrimental effect of buckling mode components of initial deflection depends to a considerable degree on plate slenderness, b/t. The bigger the slenderness, b/t, the less detrimental is the initial deflection for both longitudinal and transverse compression. Moreover, in the case of transverse compression, the detrimental effect of initial deflection does not depend on the plate aspect ratio a/b. The relative decrease in plate strength due to increase in initial deformation is the same, for instance, for $a/b = 2$ as for $a/b = 4$. By longitudinal compression on the other hand plate strength increases with a/b when the shape of the initial deflection does not conform with the plate buckling mode shape. Similar conclusions can be drawn from the recent investigation results given in (59) where ultimate strength of compressed plates, having initial deflection in the range of six plate thicknesses has been analysed.

Naturally, with other kinds of loading, e.g. bi-axial compression, bi-axial compression with lateral pressure, bi-axial compression with lateral pressure and shear, etc. which often occur in ship plating and especially in the midship section, the conclusion will differ to some degree, particularly in quantitative aspects. Yet the basic conclusion will remain the same: the detrimental effect of the initial deformation tends to decrease with the increase in structure slenderness. In such circumstances it does not seem reasonable to handle the initial deflections in such a way that the permissible value of w_o/b is independent of plate slenderness b/t.

In reference (58) a detailed analysis was made of the influence of initial deflections on the strength of stiffened plate panels subject to compression. In accordance with the results of the analysis an increase in plate initial deflection from 0.005b to 0.015b causes a decrease in the panel ultimate strength of between 5 and 10 per cent. The biggest decrease was, of course, noted in panels with the smallest slenderness. Initial deflection of this value, according to formula (19), will reduce the plate ultimate strength by 19 to 28 per cent. This applies to plates of low slenderness ($b/t = 30$) which are most sensitive to imperfections. In case of ordinary ship steel with $\sigma_y = 235$ MPa and $E = 206{,}000$ MPa this corresponds to $\beta = 1.13$.

Let us now assume that the reduction in ship shell plating ultimate strength may not exceed 19 per cent due to the existence of an initial deflection, then, for normal ship steel, on the basis of formula (19), we will obtain the permissible values of w_o/b as a function of plate slenderness b/t given in Table 6.

Table 6 Permissible values of w_o/b for $r_{IA} = 0.81$

b/t	30	60	90
w_o/b	1/200	1/111	1/22

As can be seen in Table 6 application of only one permissible value of w_o/b for all b/t, e.g. 1/200, will mean that plates of slenderness $b/t > 30$ with a fabrication deflection exceeding 1/200 will be faired although the relative reduction of their strength will be definitely lower than those of $b/t = 30$. Therefore I am not in favour of the presently used over-simplified measure of fabrication plate deflection harmfulness in the form of w_o/b or w_o where it is only dependent on plate slenderness b/t. In view of the above it is interesting to compare the recorded values of fabrication distortions with the permiss-

ible ones given in Table 6. I have made this comparison in Figure D3 where values of plate initial deflections given by the curve $r_{IA} = 0.81$ cause the same reduction in the level of plate ultimate strength (19 per cent) vs. b/t. As can be seen in the Figure the fabrication deflections occurring in practice do not constitute a significant risk for ship shell plating ultimate strength particularly for plating of $b/t \geq 60$. Therefore I propose for plating of $b/t \geq 60$ the permissible value of $w_o/b = 1/100$, whereas for that of $b/t < 60$, $w_o/b = 1/200$. These proposals apply to the midships. In ship ends where there are no loads in the plating plane I think that even $w_o/b = 1/75$ may be permitted for any value of b/t.

In Figure D3 the most recent measurement results of plate fabrication deflection in Polish shipyards are also shown. The results differ slightly from the results shown in Figure 1 of my paper. At the time of writing my paper I had at my disposal the measurement results of 236 plates only, whereas the results given in Figure D3 are based on the recordings of 862 plates.

In reference (33) of my paper the details of the theory of the process of welding distortion generation is described. In accordance with the theory the plate post-welding deflections result mainly from the transverse contraction of fillet welds used for welding stiffeners to plating. This shrinkage depends on the physical properties of weld material and the weld dimensions as well as on the welding process parameters, but is independent of the physical properties of the plate material. The plate deflections are also a function of the compressive stresses generated in the plate during welding. The magnitude of these stresses depends on the plating material yield stress as well as on the size and kind of weld, weld physical properties and welding parameters. If the value of the compressive stresses exceeds the critical stresses of the plating considerable plating deflections can result. This applies, however, to relatively thin plates, t < (6–7) mm, which are not used as ship hull essential load carrying members, so they were not subject of my study. Hence in my considerations there is no relationship between the fabrication deflections and plate material yield stress.

REFERENCES

57 CZUJKO, J.: Collapse Analysis of Plates Subjected to Biaxial Compression and Lateral Load, Department of Marine Technology, The Norwegian Institute of Technology, The University of Trondheim, November, 1983.

58 CARLSEN, C. A.: A Parametric Study of Collapse of Stiffened Plates in Compression, The Structural Engineer, Vol. 58B, No. 2, June, 1983.

59 DOW, R. S.: Effect of Damage on the Collapse and Post-Collapse Behaviour of Rectangular Steel Plates, PRADS '87, Trondheim, June, 1987.

To Mr. G. H. Sole:

The references quoted by you, (50) and (51), are very significant in view of further studies of the influence of the initial deflection geometry on the ultimate strength of plates subjected to compression. Having studied these references I share the conclusions you presented. The reference (50) indicates that initial deflections of limited local character are even more dangerous than those covering the whole plate surface. Therefore if we intended to remove initial deflections then we should consider first all those of a local nature. This is because they are always detrimental for structures, they can be spotted by the naked eye and can be effectively removed under shipyard conditions. The reference (51) indicates that in the case of a longitudinally compressed rectangular plate having an initial deflection composed of many harmonic components, then the detrimental harmonic can be practically any whose length is smaller than the plate width. But, as in shipyard conditions it is practically impossible to detect it visually, then it must be remembered that plate fairing without any control of the initial deflection geometry makes the operation even more dubious.

With regard to the effects of initial deformations on the strength of stiffeners, there are two more elements I would like to point out:

A plate is acting as effective width for a stiffener, therefore a plate initial deflection which increases its strength is also advantageous for the stiffener. On the other hand, the initial deformation of the stiffener itself having the same geometry as the plate initial deformation is generally detrimental for the stiffener, so it should be kept within specified limits. As you mentioned this applies particularly to initial deformations in the shape of one half-wave along the whole length of the stiffener. Not only in bi-axial but also in uni-axial compression along stiffeners, plate initial deflections can initiate loss of stability of the stiffeners and thus seriously deteriorate the panel strength, if plate deflection geometry contains, besides one half-wave, some higher harmonics. I had observed the phenomenon during an experimental investigation, the results of which are presented in reference (22).

To Mr. J. Frize:

I fully share your point of view that "it is better to prevent than to cure". My shipyard experience shows that it is quite possible. The point is to make proper use of present fabrication processes and the shipbuilders' knowledge and experience. "Where there is a will there is a way" says a Polish and English proverb. But whether the shipbuilders have the will depends also to a large extent on ship construction survey and that is where the Classification Societies are able to contribute as well.

To Mr. C. A. Gatt:

I am also of the opinion that fairing carried out to excess in a shipyard is generally a result of low quality workmanship and this workmanship should be verified. But there are also exceptional situations. For instance it may be necessary to apply relatively thin plates in order to reduce considerably the structure weight. Under these circumstances deformations will occur, the magnitudes of which will definitely exceed the structure deformations which one might consider to be normal.

Answering Mr. A. C. Viner's questions, on the basis of an analysis of influence of imperfections on strength of plates in compression, I proposed the following permissible fabrication ship plating deflections:

- within 0.6L amidships $w_o/b = 1/200$ for plates of $b/t < 60$ and $w_o/b = 1/100$ for $b/t \geq 60$,
- for the remaining ship parts $w_o/b = 1/75$ for all b/t.

These proposals are shown in Figure D3 against the background of statistical measurements of ship plating deflections taken in Polish shipyards and in relation to the mean value determined on the basis of standards existing in various countries ($w_o/b = 1/88$). But whether my propositions resulting from strength considerations can be accepted by ship owners for whom the essential importance, vary often, is mainly the aesthetic appearance of the ship I am not able to say. However, I would like to stress that in my opinion the Classification Society is responsible first of all for the safety and reliability of the structure and not for its aesthetics. The matters of aesthetics ought to be a question of agreement between the shipyard and the owner. If a shipyard is ready to build a ship of specified aesthetic attributes for a specified price which the Owner is ready to pay then this is not a problem for the Classification Society.

Instead of cutting out and replacing part of a deformed structure of course it would be better to have it made without excessive deformations. But in the case of very poor quality it is often unavoidable to have to replace parts of a ship's structure. This is because repeated fairing will not bring the expected result and besides introduces additional stresses to the structure which subsequently may cause cracking during service.

To Mr. M. H. P. Hembling:

The problem you mentioned has never been the subject of my research. Neither can I recommend to you any reference containing investigation results regarding effects of such deformations on panel strength, especially in respect of fatigue strength. On the basis of other investigations that I am familiar with, I can only say that where there is a large curvature there is also concentration of stresses which inevitably leads to the reduction of the structure fatigue strength. There are numerous stress concentrators in ship structures so the question arises whether the one you described is more dangerous than others or not. I am not able to answer this question either.

Considering the effects of the initial deflection curvature on the strength of plates subjected to cycling compressive loads the investigations contained in reference (60) are undoubtedly interesting. The investigations disclosed a significant reduction in ultimate strength of plates when the initial deflection curvature exceeded a certain limit value.

REFERENCES

60 KAMINSKI, M.: The Behaviour of Imperfect Plates Subjected to Cycling Compressive Loads, International Colloquium "Stability of Plates and Shell Structures", Ghent, April, 1987.

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